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EFFECT OF SURFACE AND MECHANICAL PROPERTIES ON SILICON NITRIDE BEARING ELEMENT PERFORMANCE

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This is the Final Report summarizing progress from October, 1973, through January, 1975 on a fundamental investigation of the effect of surface and mechanical properties on silicon nitride bearing element performance. This work was performed under Contract No. N00019-74-C-0168 from the Naval Air Systems Command.

The report discusses the importance of understanding the extent of surface damage caused during the manufacture of bearing surfaces on ceramics. The likely impact of the damaged surface on the usefulness of ceramics as rolling bearing materials is presented. The results of flat-washer rolling-contact fatigue tests signify the role of surface interactions in a steel-silicon nitride contact on the life of the contact. The variables evaluated include lubricant viscosity, surface finish of the silicon nitride washer, type of contact (line or point) and hardness of the steel rollers. The effect of the lower density and higher elastic modulus of silicon nitride on bearing fatigue life was analytically evaluated. The analysis shows that silicon nitride rolling elements can improve the fatigue life of very high speed bearings.

Results of friction and wear studies conducted by Professor Rabinowicz at the Massachusetts Institute of Technology are presented.

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FINAL REPORT

ON

EFFECT OF SURFACE AND MECHANICAL PROPERTIES ON SILICON NITRIDE BEARING ELEMENT PERFORMANCE

February, 1975

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EFFECT OF SURFACE AND MECHANICAL PROPERTIES ON SILICON NITRIDE BEARING ELEMENT PERFORMANCE

SUMMARY

This is the Final Report summarizing progress from October, 1973, through January, 1975 on a fundamental investigation of the effect of surface and mechanical properties on silicon nitride bearing element performance. This work was performed under Contract No. N00019-74-C-0168 from the Naval Air Systems Command.

The report is written in four parts.

Part I discusses the importance of understanding the extent of surface damage caused during the manufacture of bearing surfaces on ceramics. The likely impact of the damaged surface on the usefulness of ceramics as rolling bearing materials is presented. Results of some lapping experiments performed in the course of this program are presented.

Part II discusses the results of flat-washer rolling-contact fatigue tests conducted to understand the role of surface interactions in a steel-silicon nitride contact on the life of the contact. The variables evaluated include lubricant viscosity, surface finish of the silicon nitride washer, type of contact (line or point) and hardness of the steel rollers.

Part III discusses the results of an analytical evaluation of the effect of the lower density and higher elastic modulus of silicon nitride on bearing fatigue life. The analysis shows that silicon nitride rolling elements can improve the fatigue life of very high speed bearings.

Part IV presents the results of friction and wear studies conducted by Professor Rabinowicz at the Massachusetts Institute of Technology.

CONCLUSIONS

1. The extent of surface interaction in a rolling contact has a significant effect on the life of a contact in which one of the elements is made of silicon nitride. Improved surface finish and higher wear resistance of silicon nitride and higher hardness of the mating steel element are material parameters that lead to improved life of the contact. For rolling contacts consisting of M41 tool steel running against NC132 silicon nitride having the smoothest finish obtained in this program, a contact life to spalling failure of the steel of five to ten times Lundberg-Palmgren theoretical life was obtained. No failure of the silicon nitride occurred.

2. Theoretically, the lower density of silicon nitride is helpful in improving bearing life above 2.5×10^6 DN in spite of its higher elastic modulus. Appropriate design changes must be considered before such new materials with greatly different physical properties are put into bearing application. By optimizing the design of a typical tool steel aircraft engine mainshaft thrust bearing for silicon nitride balls, a theoretical increase in ring life of three to six times that of an all-steel bearing can be realized at high speeds.

3. Oxide abrasives appear to produce a superior surface finish on silicon nitride compared to other abrasives such as silicon carbide and Borazon used in this program. Silicon Nitride has a lower than normal abrasive wear resistance for ceramics. It exhibits high wear rate (low wear resistance) when abraded against hard abrasives such as silicon carbide or soft abrasives such as calcium pyrophosphate. This bimodal wear characteristic may be due to the duplex structure of silicon nitride consisting of hard silicon nitride grains bonded by a softer glass phase.

4. Work towards improvement and reproducitility of silicon nitride material and processing methods is required to make silicon nitride a viable bearing material. Both these parameters control the wear characteristics of silicon nitride which in turn effects the bearing life.

INTRODUCTION

Future aircraft will operate at even higher speeds and temperatures than are met today and proper bearing lubrication and life expectancy will pose an even greater challenge than it does now. The concensus is that operating speeds of 3×10^6 DN (DN = bearing bore diameter in mm times shaft speed in rpm) and temperatures in excess of 700°F will be required in the near future (1)*.

The predominant force on the balls in high-speed engine bearings is the centrifugal force which induces such high outer ring contact forces that bearing life is reduced drastically as engine bearing speeds increase. A substantial portion of this loss in contact fatigue life can be regained if light-weight balls are used.

Recent advances in ceramic processing technology have demonstrated a capability to fabricate a fully dense ceramic part. This makes it possible to take advantage of the other well recognized properties of ceramics, namely low density, high strength and high temperature structural stability. The lower density of ceramics makes it attractive for high speed applications such as very high speed jet engine bearings where the centrifugal load can become significant to the fatigue life of the bearing.

The stability and high hardness levels of ceramics at elevated temperatures have long made them appear desirable for use in bearings operating at extreme conditions. It can be expected that the use of new, stronger ceramic materials will offer improvements in high temperature properties over steels.

The large strides made in improving ceramic processing technology was possible due to the financial support of the U. S. Government (2) over the past few years. The improvements in processing technology have been particularly significant in the cases of silicon carbide (3) and silicon nitride (4). The lower elastic modulus of silicon nitride which leads to lower Hertz stresses and its greater resistance to cracking under a Hertzian load (5) makes it a more suitable material for rolling bearing applications.

*Numbers in parentheses refer to list of references at the end of this proposal.

The best grade of silicon nitride presently available in the U. S. is Noralide NC132 made by Norton Company, Worcester, Massachusetts. A list of selected physical properties are given below (6).

Density, gm/cc	3.2
Specific heat, Cal/gm°C	0.17
Coeff. of thermal expansion (25-200C),/°C	3.2×10^{-6}
Thermal conductivity, Watts/m, °K	34
Hardness, MN/m^2	22,000
Elastic modulus, MN/m^2	29×10^4
Flexural strength (4 point), MN/m^2	860
Compressive strength, MN/m^2	3440

The above list indicates that silicon nitride possesses excellent properties for structural (high strength) and wear (high hardness) type applications.

For a material to be useful in wear type applications, as in the case of bearings, it must be readily lubricable. The ability of ester and hydrocarbon base lubricants to lubricate silicon nitride has been previously demonstrated (7). The wear resistance of silicon nitride is a function of its composition, as described in this report. The rolling contact fatigue life of silicon nitride has been shown to strongly depend on the grinding technique used (8). Since wear occurs by fracture of surface asperities and wear resistance is known to be a function of composition, rolling contact fatigue life will also be dependent, to some extent, on composition.

In the following sections, results and analyses are presented to demonstrate:

a) the need to investigate ceramic surface finishing techniques for bearings,

b) the role of the surface finish of silicon nitride on the rolling contact fatigue life of a steel-silicon nitride contact; and

c) the effect of using silicon nitride balls, instead of M50 tool steel balls, on the rolling contact fatigue life of a typical high-speed jet engine mainshaft thrust ball bearing. Operating speeds were chosen to give DN values of 2, 3 and 4 million. A typical engine bearing DN value presently in use is about 1.5 million.

Wear resistance (inverse of wear coefficient) of silicon nitride was measured to be an order of magnitude lower than other ceramics. Experimental results obtained by Professor Rabinowicz explains why this may be the case.

PART 1

Effect of Finishing Method on Ceramic Surfaces

1.1 Effect of Grinding on Surface Damage and Fatigue Life of Ceramics

A structural ceramic component is manufactured by first hot pressing the powder in a mold. The hot pressed part is then rough ground, finish ground, lapped and polished to obtain the finished part. Finish grinding usually removes the damaged layer caused during the rough grinding operation. Since a very small layer of the material is removed during lapping and polishing, surface damage generated during grinding may be left in the finished part if the grinding process is not performed properly. The deleterious effect of grinding damage on the physical properties of ceramics has long been recognized. However, only a few studies, for example (8, 9, 10), have attempted to correlate grinding parameters such as grit size, grit type, coolant type and grinding speed to the extent of surface damage. It is therefore not possible to design an optimum finishing sequence at the present time. The only way to avoid the deleterious effects of grinding damage on the performance of structural parts is by performing the grinding operation as gradually as possible based upon experience. This is due to the brittleness of the ceramics which makes them significantly more sensitive to surface damage than metals. With the growing consideration of ceramics such as silicon nitride for heavily loaded applications, a good understanding of the extent and control of grinding damage becomes mandatory. A knowledge of the correlation between grinding damage and physical properties is also necessary. As far as the bearing application is concerned, the presence of grinding damage may not lower the fracture strength of silicon nitride below that of metals presently used. The lower fracture strength however, would lower wear resistance of the surface layer. The excess wear debris generated can significantly affect the life of a steel-silicon nitride contact.

As stated earlier, the field of grinding damage and its consequences has been inadequately explored. This is particularly true for alumina, silicon carbide and silicon nitride which are most popular amongst ceramics for structural applications. There are some results, however, from which the effect of surface damage produced during grinding may be deduced. The effect of finishing procedure on the rolling contact fatigue life of silicon nitride was conducted by Norton (8). Of the four sets of specimens prepared for the study, two sets are of particular interest. The first set was prepared by grinding

with 100 and 320 grit diamond followed by hand lapping with a leather strap impregnated with 6 μm diamond dust. This gave a median life in rolling contact fatigue of 31 million cycles. In the preparation of the second set the 320 grit diamond grind was omitted and lapping was performed between two cast iron plates charged with 6-8 μm diamond paste. The median life in this case was 650,000 cycles. This qualitatively indicates that the damaged surface produced during grinding with 100 grit diamond persisted in the finished specimens and gave rise to poor rolling contact fatigue life.

An insight into the extent of damage produced during grinding can be gained by examining some of the data presented by Petrovic and Jacobson (11).* In order to reduce the scatter in fracture strength of the test specimens caused by material inhomogeneities, they introduced controlled damage using a Knoop hardness indentor. Figure 1, taken from their report, shows the relation between applied load, loss in fracture strength and size of the damage zone. From their results it is found that under 2600 gm load the depth of the Knoop indentation is approximately 5.7 μm whereas the depth of the damaged zone is 68 μm. This gives an approximate ratio of 12:1 between the depth of damaged zone and depth of the dent. Using this ratio one can compute the approximate depth of the dent for the lowest load used. From Figure 1 the damaged zone is seen to be 24 μm deep. This gives a depth of 2 μm for the dent. Empirically it is known that the ratio between the peak-to-valley height to the measured surface finish is about 4:1. The 2 μm deep Knoop dent would therefore represent a surface finish of 0.5 μm AA. This gives a ratio of about 50:1 between thickness of damaged surface layer to the surface finish since grinding may be considered to be a continuous process of denting the surface. However, under dynamic conditions of grinding this ratio is expected to be smaller due to the time element involved in the generation and propagation of such damage. This type of knowledge related to the grinding of ceramics can help in optimizing the finishing process in terms of quality and cost of the finished component.

1.2 Results of Lapping Experiments

Lapping is used subsequent to grinding, primarily to improve the surface finish. A detailed investigation of the

*The purpose of their investigation was to study the effect of environment on fracture strength of silicon nitride.

lapping characteristics of silicon nitride was conducted by Gielisse (12). This work indicates that silicon carbide gives the best material removal rate. Comparative lapping experiments conducted at S K F using 600 grit (25 μm) SiC and 15 μm CBN Borazon (14) showed that Borazon gives a higher material removal rate under identical lapping conditions. Since Borazon is similar to diamond in its cutting properties, the reason for the superiority of SiC over diamond found in Gielisse's investigation is not clear. His finding that the oxides Al_2O_3, CeO and SnO_2 give the least pitted surface has been confirmed in the course of this program.

Flat washers used in this program were made from Noralide NC132 grade silicon nitride having three different levels of surface finishes.

a) As ground surface with a directional lay (Figure 5) and surface finish of 0.2-0.25 μm AA.

b) Surface lapped with 600 grit SiC having a random lay (Figure 6) and a surface finish of 0.075-0.1 μm AA.

c) Surface hand polished with 5 μm Al_2O_3 having a random lay (Figure 7) and a surface finish of 0.04-0.05 μm AA.

PART II

Flat Washer Rolling-Contact Fatigue Testing

Flat washer tests were conducted with Norton NC-132 grade silicon nitride flat washers run against either three ½-inch diameter balls or three 5 x 6 mm rollers. The tests were conducted in the S K F Flat Washer Testers shown in Figures 2, 3 and 4 and previously described (13). The tests were conducted to investigate the effect of the following parameters on the fatigue life of a contact.

a) Extent of sliding in the contact.
b) Surface finish of silicon nitride.
c) Lubricant viscosity.
d) Hardness of the steel element.

The test parameters, the resultant operating parameters and the lives obtained for the flat washer tests conducted in this program are listed in Table 1.

Comparison of fatigue lives of the roller-flat and ball-flat contacts brings out the effect of the extent of sliding on the fatigue life of steel-silicon nitride contacts. The difference in the stress level in the two contacts must be borne in mind when this comparison is made.

The effect of surface finish of the silicon nitride flat washer on contact life was studied by preparing flat washers with different surface finishes as explained in the previous section.

Contact fatigue lives obtained with MIL-L-23699 and DTE Medium Heavy lubricants are compared to determine the effect of lube viscosity on contact fatigue life. However the effect of temperature-viscosity and pressure-viscosity coefficients is also included in the above comparison. Of most significance is the lubricant film parameter which is the ratio of the lube film thickness ($h, \mu m$) to the composite surface finish ($\sigma, \mu m$) of the mating surfaces. Composite surface finish is the root-mean-square average of the surface finishes of the two mating surfaces. The method of calculating the lube film parameter ($\Lambda = h/\sigma$) is given in Appendix I. The necessary lubricant properties are given in Figure 8. Calculated values of Λ for each of the flat washer tests is also listed in Table 1.

Rollers made from AISI M50 and M41 tool steels were run to investigate the effect of hardness of the steel element on

contact fatigue life. The rollers made from the lower alloy M50 tool steel have an average hardness of R_c60 whereas those made from the higher alloy M41 tool steel have an average hardness of R_c67.

To be able to investigate all of the parameters discussed above, only a small number of tests could be conducted to establish the effect of each parameter on contact fatigue life. The life data for each set of conditions was therefore subjected to statistical analysis using Weibull statistics (14, 15) to estimate the L_{50} life for each group. Since the failing component in all but one test was the steel element, a Weibull slope of 1.3, applicable to steel, was used for this purpose. Again due to the small number of tests in each group, a 70% confidence band was calculated for each of the groups to determine if the difference in the fatigue life of any two groups is significant. The results of the statistical analysis of the life data are shown in Table 2. Theoretical lives of various contact types are given in Table 3 for comparison. The procedure for calculating the theoretical life for a steel-steel contact is shown in Appendix II. The lives of steel-silicon nitride contacts were deduced from this as shown in the Appendix.

Before discussing the flat washer test results any further, it would be desirable to understand the kinematics of a roller-flat washer test and its effect on the size and position of the wear band on the roller. A schematic diagram of the flat washer test configuration is shown in Figure 9. As seen in Figure 9a, the test flat washer and the roller are moving towards the right at the lower contact A. The velocity of a point on the surface of the flat washer varies with its radial position. The roller surface has the same velocity at any point and is equal to the velocity of the center of the roller. As seen in Figure 9b this leads to a lower flat washer velocity, relative to the roller, at the inside end of the roller and a higher flat washer velocity relative to the roller at the outside end of the roller. The relative sliding motion between the mating surfaces generates friction forces and is also responsible for generating wear on these surfaces. The magnitude of friction forces generated by sliding between the roller and the flat washer are equal and opposite at the two ends of the roller. The location of the wear band in this case would be at the center of the roller as shown in Figure 9c. An increase in the magnitude of the friction force decreases the width of the wear band whereas a decrease in the magnitude of the friction force would widen the wear band i.e., cause the roller surface

to wear more evenly. Under these conditions the wear band produced at the upper contact B would overlap that produced at contact A. The presence of a cage used to guide the rollers contributes two forces. The cage force that tends to move the roller ahead due to its own motion and the friction force due to the sliding motion between the cage and the roller. The effect of the cage force (Figure 9d) is to move the wear band formed at contact A towards the outer end of the roller and that formed by the contact B to the inner end of the roller. The widths of these wear bands need not be the same as they depend on the friction coefficients at the two contacts. The friction force between the roller and the cage on the other hand tends to move the wear band produced at contact A towards the inside of the roller whereas that caused by contact B towards the outside end of the roller. In any given test therefore the size and the position of the wear bands produced by the two contacts depend on the magnitude of the various forces acting on the roller. The results of the flat washer tests will now be discussed to highlight the effect of the various operating parameters listed earlier on the contact fatigue life. The roller-flat washer tests will be discussed first to sort out the effect of changing a test parameter on the fatigue life of a roller-flat washer contact. This is followed by a discussion of the ball-flat washer tests which demonstrate the effect of decreased sliding on the improvement of the fatigue life of the contact.

The first group of roller-silicon nitride flat washer tests was run using M50 steel rollers (R_c60), MIL-L-23699 lubricant and a silicon nitride washer having a surface finish of 0.225 μm AA (Figure 5). The surface distress suffered by the rollers in Test #3 is shown in Figure 10. This test was run at 2225 N for 3.3×10^6 cycles which is equivalent to 0.21×10^6 cycles at 4450 N, the normal load used in this program for roller-flat washer tests. The roller track on the silicon nitride flat washer developed a 0.5 μm deep groove at the end of the test. Running further tests with the grooved flat washer did not change the groove depth. The roller track surface was significantly smoother at the end of the test as shown in Figure 11. There is evidence of plastic deformation of silicon nitride surface asperities during rolling contact. It will be seen that irrespective of the initial surface finish and topography of a silicon nitride washer, its appearance after rolling contact is practically the same. To get from the initial to the final state, material is removed from the silicon nitride surface by fracture wear, as suggested by Rabinowicz in Part IV. The greater the amount of change in

the surface the larger the quantity of wear debris produced. Occurrence of fracture wear also leads to grooving of the flat washer surface. Elimination of fracture wear stops grooving of the silicon nitride washer as will be shown later.

In Test Group #1 the operating conditions lead to a lube film parameter range of 0.37-0.54 (Table 1). As seen from Figure 12 the available lube film is too thin to provide adequate separation between the mating surfaces. The sharp asperities on the silicon nitride surface initiate the damage on the roller surface. When the asperities are worn away, the wear debris in the oil continues to damage the roller surface. The median unbiased maximum likelihood estimate of the L_{50} life (henceforth referred to as the median L_{50} life) of a roller-flat washer contact in Group #1 is 0.55×10^6 cycles (Table 2) which is significantly lower than the theoretical value of 16×10^6 cycles (Table 3). The deleterious effect of the 0.225 μm AA finish on the silicon nitride washer on the fatigue life of the contact is quite evident.

In Group #2, M41 steel rollers (R_c67) were used under operating conditions similar to Group #1. In spite of a lower lube film parameter range of 0.27-0.31, the median L_{50} life increased to 2.93×10^6 cycles. Although an improvement, the fatigue life of the contact is still lower than the theoretical value. The extent of roller surface damage is practically the same (Figure 13). The time required to damage the M41 steel rollers by the same amount is increased due to its higher hardness.

The position of the wear band in both the above groups was in the center of the roller. The width of the wear band on M50 rollers was greater than on the M41 steel rollers. Since the coefficient of friction in the contact for the two materials is not expected to be different, the wider wear band on the M50 rollers is merely indicative of the lower wear resistance of M50 steel compared to the harder M41 steel.

In Group #3, the lubricant was changed from the lower viscosity MIL-L-23699 to a higher viscosity DTE Medium Heavy oil. This leads to a higher lube film parameter range of 0.49-0.64. The median L_{50} life of the roller-flat washer contact increased to 4.81×10^6 cycles. The type of damage remains the same but the extent is considerably reduced (Figure 14). The presence of two wear bands on the roller surface indicates that the cage forces are more predominant than the friction forces in the contact. The formation of a 0.5 μm

deep groove on the silicon nitride flat washer was observed in this test group also.

Group #4 consisted of one test using M41 steel rollers instead of M50 steel rollers with other operating parameters same as Group #3. With a lube film parameter of 0.52 an L_{50} life of 9 x 10^6 cycles was obtained. The increase in life can be attributed to the higher hardness of the M41 steel rollers.

The difference in operating parameters between Groups 3 and 5 is the improvement in the surface finish of the silicon nitride flat washer from 0.225 μm AA to 0.1 μm AA. The lube film parameters for the two tests conducted were 0.85 and 1.34 which are approximately twice as large as those in the previous groups. The median L_{50} contact life for the group is calculated to be 16.5 x 10^6 cycles. Considering the small sample size used, the absolute value of the life should be used with caution. A comparison of the 70% confidence interval of 2.26-8.32 x 10^6 cycles of Group #3 with an interval of 5.64-9862 x 10^6 cycles does suggest the positive effect of better flat washer surface finish used in Group #5 on the fatigue life of the contact. Again the type of roller surface damage is the same as before (Figure 15). The time to failure is increased due to the reduced surface interaction between the mating surfaces. The post test condition of the roller track on the silicon nitride washer was the same as shown in Figure 11. The groove depth on the roller track of the 0.1 μm AA flat washer was the same (0.5 μm) as on the rougher washer. This indicates the persistance of fracture wear at the 0.1 μm AA surface finish level. Groove formation on a silicon nitride bearing surface during rolling contact was observed by other workers (16).

The next set of roller-flat washer tests were run in Group #8 using a silicon nitride flat washer surface finish of 0.05 μm AA. The lube film parameter range was 1.83-2.44. The median L_{50} life was 6.06 x 10^6 cycles. This is poor compared to the life of 16.58 x 10^6 cycles obtained for Group #5. This discrepancy is however likely to be due to the small sample size of two for Group #5 with only one failure. A more reasonable comparison can be made with Group #3 which has the same sample size and number of failures. The improvement in flat washer surface finish has improved the contact fatigue life in Group #8. The improvement however is marginal and the fatigue life is lower than the theoretically expected value of 16 x 10^6

cycles. Wear on the M50 steel roller surface is evenly distributed as seen in Figure 16. The roller track on the flat washer was not grooved. This signifies the elimination of fracture wear of silicon nitride at this surface finish level.

The use of M41 steel rollers against 0.05 μm AA finish silicon nitride flat washer in Group #9 leads to a significant improvement in the contact life. The median L_{50} life is 151.8×10^6 cycles. In spite of the small sample size used in this group the increase in fatigue life is large enough to indicate that the higher hardness of the M41 steel rollers makes a significant contribution to the improvement of the roller-flat washer contact life. The lower 70% confidence limit of 71.34×10^6 cycles is significantly greater than the theoretical L_{50} life of 16×10^6 cycles. The appearance of the roller surface (Figure 17) from Test #28, suspended at 72×10^6 cycles, shows only minimal damage. The roller track on the flat washer was ungrooved. Its appearance after test is shown in Figure 18.

The above discussion on the roller-flat washer tests indicates that improved surface finish of the silicon nitride flat washer, higher lube viscosity and higher hardness of the steel element lead to improved contact fatigue life. Of the three parameters, the hardness of the steel element has the largest effect on life improvement once the surface finish of the silicon nitride washer is fine enough to prevent fracture wear of the material.

The ball-flat washer tests were conducted to determine the effect of reduced sliding on the contact fatigue life. These tests are divided into two groups. Those tests run with a new silicon nitride flat washer (Group #6) and those run with a silicon nitride flat washer previously used in a ball-flat washer test (Group #7). Since a ball-flat contact is a point contact the Hertz stresses in the contact are higher than in a roller-flat contact at the same load. In the present program an applied load of 3560 N was used for the ball-flat tests. This produced a maximum Hertz stress in the contact of 4690 N/mm^2. The maximum Hertz stress in the roller-flat washer contact was only 2100 N/mm^2 produced by an applied load of 4450 N.

The median L_{50} life of a contact in Group #6 was 39.83×10^6 cycles. This compares favorably with the theoretical value of 13.23×10^6 cycles. The balls used in this group

did not possess a visible wear band as seen in Figure 19. At the end of the test the ball track on the silicon nitride flat washer contained a 1 μm deep groove.

The silicon nitride flat washers grooved in the above tests were used for tests in Group #7. The presence of the groove increased the extent of sliding in the ball-flat washer contact. The increased sliding in the contact in the presence of high Hertz stresses decreased the median L_{50} life of the contact to 6.0×10^6 cycles. The balls used in these tests had a visible wear track. The surface material in the wear track was distressed and smeared. This was not found to be the case for balls used in Group #6. The surface damage on balls used in Test #19 of Group #7 is shown in Figure 20.

The only silicon nitride flat washer failure occurred in Test #19. Examination of the ball track on the flat washer revealed that spalling of the silicon nitride flat washer is initiated by the formation of Hertz cracks during rolling contact. Figure 21 shows the Hertz cracks formed in the ball track and the spall formation from such cracks.

The groove formation in the silicon nitride ball track was mathematically analyzed as shown in Appendix III. A value of 2×10^{-6} for the adhesive wear coefficient, determined by Rabinowicz, was used for the NC 132 grade silicon nitride used in these tests. The result shows that a groove 1 μm deep would form in the ball track in about 0.33×10^6 cycles. If this is the case, then the balls run against a new washer must live as long as they do because of the development of an exact conformity between the ball and the groove. New balls used on a grooved washer would not conform as well and suffer excessive surface damage which in turn would lead to reduced life of the balls. Since in practice balls may need to be replaced in a bearing, it is imperative that the wear of the silicon nitride be reduced as much as possible. This implies use of proper composition and surface finish of the silicon nitride material. It also implies that running-in of bearing components at a lower load for a short time before putting the bearing into use, would develop the necessary conformity between the mating surfaces and prolong the life of a bearing.

Test #25 was run using silicon intride balls and flat washer. Since in practice a change in bearing material does not permit one to change the bearing load, this test was also conducted at an applied load of 3560 N. The higher modulus

of the silicon nitride-silicon nitride contact leads to a higher maximum Hertz stress of 5360 N/mm^2. This Hertz stress was high enough to cause bulk deformation of the ball track. The life of the test was 4.2 x 10^6 cycles. Due consideration must therefore be given to a bearing design intending to use silicon nitride, to account for the effects of the higher modulus of the material.

Test #26 was run with AISI 52100 steel washer and silicon nitride balls. The purpose was to determine the effect of the location of silicon nitride on contact fatigue life. The life of the test was 2.55 x 10^6 cycles which indicated that the position of silicon nitride in the contact did not have a significant effect on the contact fatigue life.

In conclusion one can list the following measures as being important to prolonging the life of a steel-silicon nitride contact.

1) Reduction of the extent of sliding in the contact.

2) Improvement in the surface finish of the silicon nitride element.

3) Use of as high a viscosity lubricant as possible in an application.

4) Increase in the hardness of the steel element.

Measure 1 reduces the extent of damage caused when mating surfaces interact. Measures 2 and 3 tend to reduce the extent of surface interaction between mating surfaces and hence the severity of surface damage. Measure 4 increases the resistance of the steel element to surface damage during rolling contact.

PART III

Analysis of the Effect of Silicon Nitride on Fatigue Life of High Speed Bearings

Ceramic materials have a lower density and a higher elastic modulus compared to metals. Their use as rolling elements in bearings produces different values of contact stress, stressed volume and centrifugal force than those for steel for a given set of operating conditions. It is important to know the effect of these changes on bearing life.

The analysis was conducted in two steps. In the first analysis only the effect of centrifugal load was considered. The second analysis was conducted for a 120mm bore angular contact ball bearing used in jet engines. The combined effect of applied and centrifugal loads on bearing life was evaluated with the help of an S K F Computer Program, AT74Y001. The analyses assume the applicability of the Lundberg-Palmgren (17, 18) Law to the life of bearings containing ceramic rolling elements.

The effect of density and modulus change of the rolling element on fatigue life under pure centrifugal load can be evaluated with the help of the probability of failure relationships given below.

$$\log\frac{1}{S} = Q^{\frac{10}{3}} \cdot E'^{7} \cdot N^{\frac{10}{9}} \qquad \text{point contact}$$

$$\log\frac{1}{S} = Q^{\frac{9}{2}} \cdot E'^{5.2} \cdot N^{\frac{9}{8}} \qquad \text{line contact}$$

where;

S = probability of survival
Q = load
E' = reduced modulus
N = stress cycles

For a given value of survival probability

$$N \sim Q^{-3} \cdot E'^{-6.3} \qquad \text{point contact}$$

$$N \sim Q^{-4} \cdot E'^{-5.2} \qquad \text{line contact}$$

Since a purely centrifugal load is proportional to the material density (ρ) we have;

$$N \sim \rho^{-3} \cdot E'^{-6.3} \qquad \text{point contact}$$

$$N \sim \rho^{-4} \cdot E'^{-5.2} \qquad \text{line contact}$$

If N_1 and N_2 are the lives of bearings containing steel and silicon nitride rolling elements respectively, then;

Point Contact

$$\frac{N_2}{N_1} = \left(\frac{\rho_1}{\rho_2}\right)^3 \cdot \left(\frac{E_1'}{E_2'}\right)^{6.3}$$

$$= \frac{1}{(0.405)^3 \, (1.2)^{6.3}}$$

$$= 4.8$$

Line Contact

$$\frac{N_2}{N_1} = \left(\frac{\rho_1}{\rho_2}\right)^4 \cdot \left(\frac{E_1'}{E_2'}\right)^{5.2}$$

$$= \frac{1}{(0.405)^4 \, (1.2)^{5.2}}$$

$$= 14.4$$

Under purely centrifugal loading silicon nitride is seen to improve the bearing life. The improvement factor is greater in the case of roller bearings (line contact) than in the case of ball bearings (point contact).

In a steel-silicon nitride contact the strength of silicon nitride and hence its fatigue life is significantly greater than that of steel. Steel-silicon nitride flat washer tests conducted in this program have shown that the failure of silicon nitride flat washers is rare. Discounting the probability of failure of silicon nitride in a steel-silicon nitride contact increases the life improvement factors as follows;

Point Contact

$$\frac{N_2'}{N_1} = \left(\frac{N_2}{N_1}\right) \cdot 2^{9/10} = 8.9$$

Line Contact

$$\frac{N_2'}{N_1} = \left(\frac{N_2}{N_1}\right) \cdot 2^{8/9} = 26.8$$

The effect of lower density and higher modulus of silicon nitride, compared to steel, on the fatigue life of a 120mm bore angular contact bearing was evaluated with the help of an S K F bearing dynamics program, AT74Y001.

In a similar previous analysis (19) it was found that the low density of silicon nitride does not improve the fatigue life of a high speed bearing. This was attributed to the increase in contact stress due to its higher modulus of elasticity. The analysis however was conducted using the same free operating contact angle for bearing containing steel and silicon nitride balls. The difference in density leads to significantly different contact angles during bearing operation. The life data therefore reflects the combined effect of material properties and contact angle.

To alleviate this problem, operating contact angles of 21° and 18.84°, for steel and silicon nitride balls respectively, were derived by trial and error, to give the same inner ring contact angle of 30° under operating conditions of 13,350 N thrust load and 3 x 10^6 DN. The data obtained in this manner reflects only the effect of material properties on bearing fatigue life. Inner ring conformity (ratio of inner ring to ball curvature) of 0.54 was used for steel balls and 0.52 for silicon nitride balls in conformity with the analysis conducted in (19), to obtain about the same shape of the contact ellipse and thus heat generation rate for the two ball materials. The results of the complete analysis for shaft speeds of 2, 3 and 4 x 10^6 DN and thrust loads of 4480 and 13,350 N, is given in Table 4.

Examination of the results reveals the following important points.

1) Under a thrust load of 13,350 N, the fatigue life of bearings containing silicon nitride balls improves over one containing steel balls at about 2.5 x 10^6 DN.

2) As the applied load decreases, the predominance of centrifugal load on fatigue life increases. The cross-over between fatigue lives of steel and silicon nitride bearings would therefore occur at a lower speed.

3) In spite of the tighter conformity used for the bearing containing silicon nitride balls, the total contact area between the ball and the rings is smaller than in the case of the steel balls. Heat generation in a bearing is a function of the contact area, amount of sliding in the contact and friction coefficient. From previous lubrication studies (7) the friction coefficients for the two contacts can be considered to be the same. The smaller contact area in the bearing containing silicon nitride balls may, therefore, generate less heat.

The results show that silicon nitride rolling elements can improve the fatigue life of high speed ball bearings. As seen from the case of pure centrifugal loading, this improvement would be even greater for roller bearings. Furthermore, if future real life high speed tests verify the initial results in this report that silicon nitride rolling elements do not fail, a further life improvement can result.

PART IV

Friction and Wear Studies

The friction and wear properties of silicon nitride under different operating conditions were conducted by Professor Rabinowicz at the Massachusetts Institute of Technology.

The investigations have revealed that the lubricated sliding friction coefficient in a steel-silicon nitride contact is similar in magnitude to a steel-steel contact. The dry sliding friction coefficient for a steel-silicon nitride contact (0.17) is only slightly higher than the corresponding value for the lubricated contact (0.12-0.13). The dry friction coefficient for a steel-steel contact is 0.54. Silicon nitride therefore has good potential for applications in marginal or lost lubrication conditions.

The wear resistance of silicon nitride is found to be a function of composition. Silicon nitride also exhibits a bimodal wear behavior.

The results and discussion by friction and wear studies conducted by Professor Rabinowicz are presented in Appendix IV.

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APPENDIX I

Calculation of EHD Lubricant Film Thickness for Flat Washer Test Conditions

C. Lubricant Film Thickness in a Ball-Flat Contact

Applied load = 800 lbs.
No. of balls = 3
Test Speed = 3400 rpm
Lube = MIL-L-23699

$$h_o = 0.87R\left(\frac{\eta \alpha V}{R}\right)^{0.74}\left(\frac{Q}{E'R^2}\right)^{-0.08}$$

where

R = ball radius (in.) = 0.25
η = absolute viscosity (lb-sec/in^2) = 7.3×10^{-7}
α = pressure-viscosity coefficient (in^2/lb) = 1.1×10^{-4}
V = entrainment velocity (in/sec) = 180
Q = load/ball (lbs) = $\frac{800}{3}$

E' = reduced modulus (psi) = 39×10^6

$$\left(\frac{\eta \alpha V}{R}\right)^{0.74} = \left(\frac{7.3 \times 10^{-7} \times 1.1 \times 10^{-4} \times 180}{0.25}\right)^{0.74}$$

$$= \underline{4.4 \times 10^{-6}}$$

$$\left(\frac{Q}{E'R^2}\right)^{-0.08} = \left(\frac{800}{3} \times \frac{1}{39\times10^6} \quad \times \frac{1}{0.0625}\right)^{-0.08}$$

$$= \underline{2.08}$$

$$h_o = 0.87 \times 0.25 \times 4.4 \times 10^{-6} \times 2.08$$

$$= 2\ \mu\text{in. } (0.05\ \mu\text{m})$$

APPENDIX I (Continued)

D. Lubricant Film Thickness in a Roller-Flat Contact

Applied load = 1000 lbs.
No. of rollers = 3
Test speed = 3400 rpm
Lube = MIL-L-23699

$$h_o = 1.95R\left(\frac{\eta\alpha V}{R}\right)^{0.727}\left(\frac{P}{E'R}\right)^{0.091}$$

where P = load per unit length of roller

$$= \frac{1000}{3} \times \frac{1}{0.228} = 1485 \text{ lb/in.}$$

R = radius of the roller = 0.1 in

$$\left(\frac{\eta\alpha V}{R}\right)^{0.727} = \left(\frac{7.3 \times 10^{-7} \times 1.1 \times 10^{-4} \times 180}{0.1}\right)^{0.727} = \underline{1.067 \times 10^{-5}}$$

$$\left(\frac{P}{E'R}\right)^{0.091} = \left(\frac{1485}{39 \times 10^6 \times 0.1}\right)^{0.091} = \underline{0.442}$$

$$h_o = 1.95 \times 0.1 \times 1.067 \times 10^{-5} \times 0.442$$

$$= \underline{1.5\ \mu\text{in}\ (0.038\ \mu\text{m})}$$

E. For DTE Medium Heavy lubricant @ 210°F

$\eta = 11.6 \times 10^{-7}$ lb-sec/in^2
$\alpha = 0.99 \times 10^{-4}$ in^2/lb

F-4186A R100

APPENDIX I (Continued)

The values for the lubricant film thicknesses for the two cases with DTE Medium Heavy oil are;

Ball-Flat: $h_o = 2.6\ \mu in (0.065\ \mu m)$

Roller-Flat: $h_o = 2.0\ \mu in (0.05\ \mu m)$

Under identical test conditions DTE Medium Heavy provides slightly better lubrication than MIL-L-23699. This in turn would lead to an improvement in the life of the rolling elements, which was observed.

APPENDIX II

Flat Washer Life Calculations

A. For Roller-Flat Configuration Using Three 5 x 6 mm Rollers

$$L_{10} = (C/P)^{10/3}$$

where P = applied load
C = capacity of the bearing
$C = g_c C_i$

$$g_c = \{1 + (\frac{C_i}{C_e})^w\}^{-1/w}$$

for 90° Thrust Roller Bearings

$$\frac{C_i}{C_e} = 1$$

$$\rightarrow g_c = \{1 + (1)^{9/2}\}^{-2/9} = 2^{-2/9} = 0.89$$

$$C_i = 1.15C$$

$$C = f_e f_a (i\ell_a)^{7/9}\, z^{3/4}\, D_a^{29/37}$$

i=rows of contacts
z=number of contacts
D_a=roller diameter
r_i=inner ring radius
r_e=outer ring radius
R_e=corner radius on roller

when $f_e = \upsilon\lambda g_c f_1 f_2$
f_a=1

$\ell_a = (\ell - 2R_e) = 6 - 2(0.3) = 5.4$; $(5.4)^{7/9} = 3.73$ mm

D_a=5mm; $D_a^{29/37} = (5)^{29/27} = 5.68$

z=3; $z^{3/4} = 2.28$

$f_i = 56.2\eta = 56.2(1 - k\sin\alpha) = 52.2(1-0.15) = 44.4$

$f_2 = \gamma^{2/9} = (D_o/d_m)^{2/9} = (5/57)^{2/9} = (0.088)^{0.822} = 0.584$

$C = \upsilon\lambda(44.4 \times 0.584 \times 0.87)\ 3.73 \times 2.28 \times 0.568$

$= \upsilon\lambda(1110 \text{ kg})$ or $\upsilon\lambda(2430 \text{ lb.})$

APPENDIX II (CON'T)

from Acta it appears that a value of $\lambda \upsilon \eta = 0.60$ is normal

$$\rightarrow \lambda\upsilon(0.85) = 0.60 \; : \; \lambda\upsilon = \frac{0.60}{0.85} = 0.7$$

$C = 0.7(8430) = 1700$ lbs.

$C_i = 1700\ (1.15) \approx 1950$ lbs.

Under a thrust load of 1000 lbs. (4450 N)

Life of the assembly:

$$L_{10} = \left\{\frac{C}{\upsilon_p}\right\}^4 = \left\{\frac{1700}{1.36\text{x}1000}\right\}^4 = \left\{\frac{1700}{1360}\right\}^4 = (1.25)^4 = 2.44 \text{ mill. revs.}$$
$$= 3.66 \text{ mill. cycles}$$

Life of one of the Roller-Flat contacts:

$$L_{10_i} = \left\{\frac{C_i}{\upsilon_p}\right\}^4 = \left\{\frac{1950}{1.36\cdot 1000}\right\}^4 = 4.18 \text{ mill. revs.}$$
$$= 6.27 \text{ mill. cycles}$$

Under a thrust load of 1200 lbs. (5430 N)

$$L_{10} = \left\{\frac{1700}{1.36\text{x}1200}\right\}^4 = \left\{\frac{1700}{1630}\right\}^4 = (1.04)^4 = 1.5 \text{ mill. revs.}$$
$$= 3.02 \text{ mill cycles}$$

$$L_{10_i} = \left\{\frac{1950}{1630}\right\}^4 = (1.19)^4 = 2.01 \text{ mill. revs.}$$
$$= 3.02 \text{ mill. cycles}$$

Under a thrust load of 800 lbs. (3560 N)

$$L_{10} = \left\{\frac{1700}{800\text{x}1.36}\right\}^4 = \left\{\frac{1700}{1630}\right\}^4 = (1.56)^4 = 5.9 \text{ mill. revs.}$$
$$= 8.85 \text{ mill. cycles}$$

$$L_{10_i} = \left\{\frac{1950}{1090}\right\}^4 = (1.79)^4 = 10.3 \text{ mill. revs.}$$
$$= 15.45 \text{ mill. cycles}$$

APPENDIX II (CON'T)

Flat Washer Life Calculation

B. For Ball-Flat Configuration Using Three 12.7 mm (½") Balls

$L_{10} = (C/P)^3$ where P = applied load and

C = capacity of the bearing

$= f_c f_a \, i^{0.7} \, Z^{2/3} \quad F(Da)$

$$f_c = \lambda g_c f_1 f_2 \left\{ \frac{2r_i}{2r_i - D_a} \right\}$$

$$g_c = \left\{ 1 + \left(\frac{C_i}{C_e}\right)^{10/3} \right\}^{-0.3}$$

i=rows of contacts
Z=number of contacts
D_a=ball diameter
r_i=inner ring radius
r_e=outer ring radius

$$\frac{C_i}{C_e} = f_3 \left\{ \frac{r_i}{r_e} \; \frac{2r_e - D_a}{2r_i - D_a} \right\}^{0.41}$$

for a 90° thrust bearing;

$f_a = 1$ $\qquad d_m$ = pitch diameter

$f_1 = 10$ $\qquad \lambda = 0.60$

$f_2 = 0.3$ where $= \frac{D_a}{d_m} = 0.48$

$f_3 = 1$

For $D_a \leq 25.4$ mm; $F(D_a) = D_a^{1.8}$

For the case under consideration:

$i = 1$ $\qquad D_a = 12.7$ mm $\qquad r_i = r_e = \infty$

$Z = 3$ $\qquad d_m = 57$ mm

APPENDIX II (CON'T)

Substituting and rearranging:

$$\frac{C_i}{C_e} = \left\{ \frac{2r_i r_e - r_i D_a}{2r_i r_e - r_e D_a} \right\}$$

$$= \left\{ \frac{2 - D_a/r_e}{2 - D_a/r_i} \cdot \frac{1/r_i r_e}{1/r_i r_e} \right\}$$

Since $r_i = r_e = \infty$; $D_a/r = 0$

$$C_i/C_e = 1$$

$$g_c = \{1 + (1)^{10/3}\}^{-0.3}$$

$$= (2)^{-0.3} = 1/1.231$$

$$= 0.812$$

$$f_c = (0.60)(0.812)(10 \cdot \frac{0.48}{0.60})(\frac{12.7}{57})^{0.3} \left\{ \frac{1/r_i}{1/r_i} \cdot \frac{2}{2-D_a/r_i} \right\}$$

$$= 3.90(0.233)^{0.3} = 3.90\ (0.637)$$

$$= 2.48$$

$$C = 2.48(1)(1)^{0.7}(3)^{2/3}(12.7)^{1.8}$$

$$= 2.48(2.08)97$$

$$= 500 \text{ Kg or } 1100 \text{ lbs.}$$

Since

$$C = g_c\, C_i$$

$$C_i = 1100 \text{ lbs.}/0.812$$

$$= 1355 \text{ lbs.}$$

Under a thrust load of 600 lbs. (2670 N)

Life of the assembly:

$$L_{10} = (1100/600)^3 = 6.16 \text{ mill. revs.}$$
$$= 9.24 \text{ mill. cycles}$$

APPENDIX II (CON'T)

Life of one Ball-Flat Contact:

$$L_{10} = (1355/600)^3 = 11.5 \text{ mill. revs.}$$
$$= 17.25 \text{ mill. cycles}$$

Under a thrust load of 800 lbs. (3560 N)

Life of the assembly:

$$L_{10} = (1100/800)^3 = 2.6 \text{ mill. revs.}$$
$$= 3.9 \text{ mill. cycles}$$

for a Ball-Flat contact:

$$L_{10} = (1355/800)^3 = 4.9 \text{ mill. revs.}$$
$$= 7.35 \text{ mill. cycles}$$

The relationships between reduced modulus and contact stress and fatigue life for line and point contacts are given below.

	Line Contact	Point Contact
Stress	$(E')^{1/2}$	$(E')^{2/3}$
L_{10} Life	$(\text{Stress})^{-8}$	$(\text{Stress})^{-9}$

The effect of reduced modulus on contact fatigue life is shown in the table below.

Contact	Combination	Stress Factor	App. Load (N)	L_{10} Life 10^6 Cycles
1. Roller-Flat	steel-steel	1	4450	6.27
2. Roller-Flat	steel-Si_3N_4	1.09	4450	3.2
3. Roller-Flat	Si_3N_4-Si_3N_4	1.21	4450	1.38
4. Ball-Flat	steel-steel	1	4690	7.35
5. Ball-Flat	steel-Si_3N_4	1.12	4690	2.65
6. Ball-Flat	Si_3N_4-Si_3N_4	1.28	4690	0.81

APPENDIX III

I. Calculation of the number of cycles required to wear a 1 μm deep groove in the ball track of a silicon nitride flat washer:

The sliding distance per unit rolling distance at a point r from the center of a ball-flat contact

$$= \frac{2r}{D_o} \cdot \frac{\Omega}{\omega} \quad \text{(Ref. 18)}$$

where D_o = ball diameter

and Ω/ω = slide-to-roll ratio

= 0.445 for the present tests

The load at a point r from the center of the contact = stress x area

$$= \sigma_{max} \left(1 - \frac{r^2}{a^2}\right)^{1/2} \times 2\pi r\, dr$$

where σ_{max} = max. Hertz stress

and a = semi-major axis of the contact

= 0.4 mm

The product of load x sliding distance at a point r from the center of the contact

$$= \frac{4\pi\sigma_{max}}{D_o} \cdot \frac{\Omega}{\omega} \cdot r^2 \left(1 - \frac{r^2}{a^2}\right)^{1/2} dr \qquad (1)$$

Putting $A = \frac{4\pi\sigma_{max}}{D_o} \cdot \frac{\Omega}{\omega}$ and $r = ax$ equation (1) reduces to

$$A a^3 x^2 (1-x^2)^{1/2}\, dx \qquad (2)$$

Integrating equation (2) over the width of the contact gives;

$$\int \text{load x sliding distance} = A a^3 \int_0^1 x^2 (1-x^2)^{1/2}\, dx$$

$$= A a^3 \left[\frac{x}{4}(1-x^2)^{3/2} + \frac{1}{8}\left\{ x(1-x^2)^{1/2} + \arcsin x \right\} \right]_0^1$$

APPENDIX III (Cont.)

$\sim c \cdot 2Aa^3$

For $\quad = 4690\ N/mm^2$
$a = 0.4\ mm$
$D_0 = 12.45\ mm$
$\Omega/\omega = 0.445$

load x sliding distance $\sim$ 27 N mm/mm

Wear coefficient $k = \dfrac{3 \times \text{hardness} \times \text{volume}}{\text{load} \times \text{sliding distance}} = 2 \times 10^{-6}$

The length of the ball track = 57 mm
Volume of a V-groove 1 m deep in the ball track = $72 \times 10^{-3}\ mm^3$
Hardness of silicon nitride = $22000\ N/mm^2$
Average sliding distance x load/cycle = 57 x 27 x 1.5
= 7252.4 N mm/mm

No. of cycles to produce a 1 m deep groove

$$N = \frac{3 \times 22000 \times 72 \times 10^{-3}}{2 \times 10^{-6} \times 7252.4}$$

0.33×10^6 cycles

II. Position of Maximum Adhesive Wear Rate Across Ball Track

This is obtained by maximizing the variable function $y = x^2(1-x^2)^{\frac{1}{2}}$ in equation (2) above.

$$dy/dx = 2x(1-x^2)^{1/2} - x^3(1-x^2)^{-1/2} = 0$$

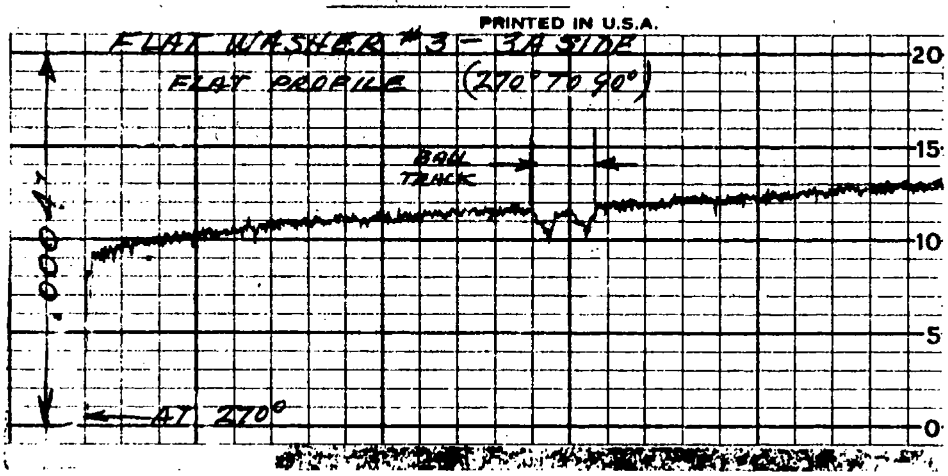
$$2x(1-x^2) - x^3 = 0$$

$$x(3x^2 - 2) = 0$$

$$x = 0 \text{ or } \pm 0.82$$

APPENDIX III (Cont.)

Since there is no sliding at x = 0, maximum wear is expected to occur at x = 0.82 i.e. r = 0.82 a. In the present case where a = 0.4 mm, = 0.33 mm. The experimentally measured wear pattern shown in the tracing below agrees reasonably well with the calculation.



APPENDIX IV

FRICTION AND WEAR TESTS WITH SILICON NITRIDE

by Professor E. Rabinowicz, MIT

Sliding tests on lubricated silicon nitride at elevated temperatures

Introduction

One of the projected uses of silicon nitride is as a ball material for high load, high speed ball bearings. In this application, the bearings are operated in a lubricated condition, and high interfacial temperatures are reached. Accordingly, to help judge the suitability of silicon nitride for this application, we have measured the friction and wear behavior of lubricated silicon nitride at elevated temperatures. Since M50 steel is the material currently used in this application, we have also carried out tests with this material to provide a basis for comparison.

Actually, there are other factors which make high temperature sliding tests of lubricated silicon nitride of interest. As earlier studies (reported last year) have shown silicon nitride is rather effectively lubricated by most typical lubricants, distinctly better than are other ceramic materials. This suggests that silicon nitride has a surface with high surface energy, making it possible for lubricants to adhere strongly to it. It is known that raising the temperature generally leads to the desorption of lubricants with an increase of friction. Thus, a study of the frictional behavior of silicon nitride at elevated temperatures allows us to estimate how strong the adhesion between lubricants and silicon nitride surfaces really is.

At the same time as high temperature tests on silicon nitrides were being carried out, it was decided to evaluate another ceramic material to see how effectively it could be lubricated. The material chosen was aluminum oxide, and we selected it rather than some other ceramic because it had not been evaluated in the room temperature tests carried out last year, so that

the results would be especially informative.

Experimental details

The elevated temperature sliding tests were carried out using a three-pin on a flat sliding geometry. The apparatus used is shown in schematic form in figure one, and it will be seen that the sliding surfaces are inside a metallurgical furnace, thus allowing testing to be carried out to very high temperatures. In these tests, the top temperature used was 250°C, since above this temperature lubricant deterioration was rapid and severe.

For the friction tests, the sliding was started at room temperature, and the friction was monitored as the temperature was gradually increased. For the wear tests, a constant temperature was maintained, and the loss of weight was measured after sliding at a speed of 5 cm/sec for a period varying from 2 to 24 hours. The normal load per pin, each pin being a rod of 1/4" diameter with a conical end, was 550 gm.

Results

The results obtained are shown in figures 2-10. As will be seen, there was an increase of friction with temperatures in all cases, and this is agreement with the typically observed behavior of most lubricants on most surfaces. On silicon nitride, the lubricants MIL-L-23699 and Terresso V78, which were found to be better than the other liquids in previous testing, were again found to be best in this series of tests.

Silicon nitride was found in general to give somewhat higher friction than the M50 steel, but the difference was relatively small in all cases.

As to the aluminum oxide, when tested with each lubricant it gave higher friction than did the silicon nitride evaluated with the same lubricant (except that with the V_{con} fluid at a temperature above 175°C, the aluminum

oxide gave lower friction). The differences were generally most pronounced in the range room temperature to 100°C.

Discussion

These results are of considerable interest. It is clear that the low friction coefficient of Si_3N_4 is generally maintained to very high temperatures. Thus, for Si_3N_4 on Si_3N_4 lubricated by the MIL-L-23699 diester lubricant, the friction coefficient remains below 0.15 at temperatures up to 200°C, only slightly inferior to the combination M50 in M50 lubricated by MIL-L-23699 (friction coefficient 0.13 at 200°C). The results with Teresso V78 mineral oil are very similar. In general, the results suggest that in switching from M50 steel surfaces to silicon nitride surfaces in high speed, high rolling contact bearings, it is unlikely that failures caused by adhesive wear phenomena will be encountered.

The other inorganic solid tested, namely aluminum oxide, gave friction values with almost all the lubricants that were higher than those obtained with silicon nitride. This confirms the results obtained last year, which suggested that silicon nitride gives exceptionally low friction, especially in the presence of boundary lubricants.

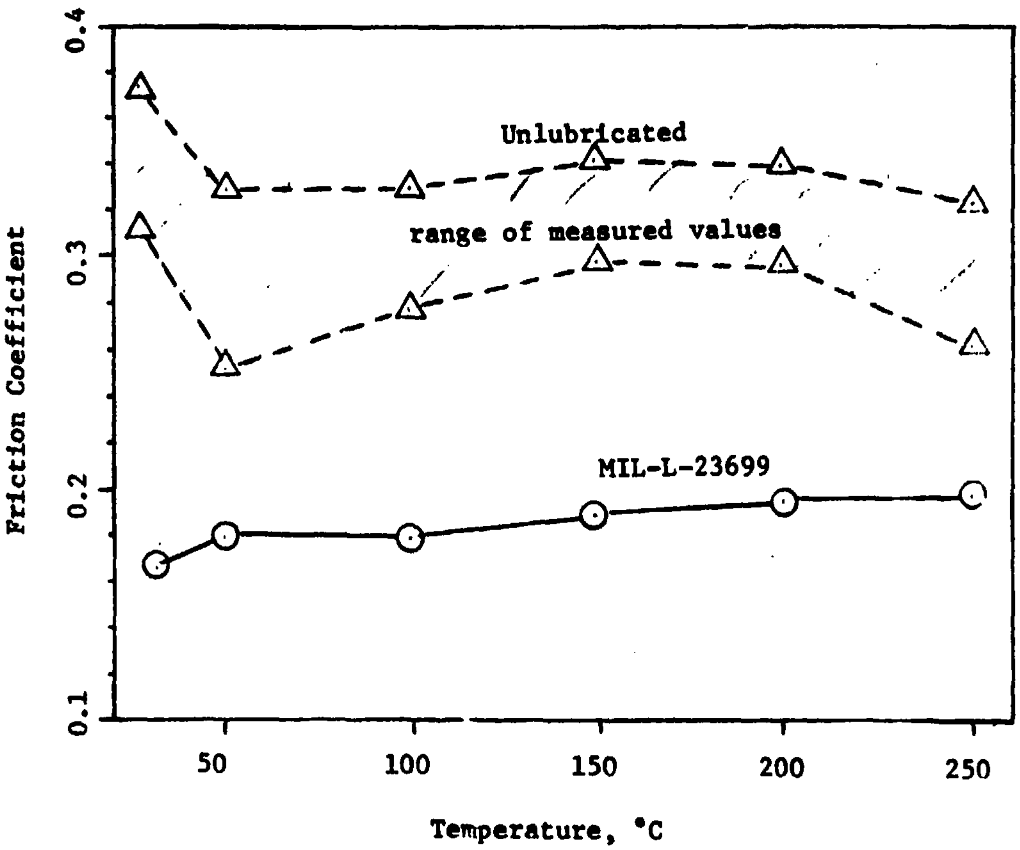


Figure 8. Effect of temperature on the friction of aluminum oxide, both unlubricated and lubricated by MIL-L-23699.

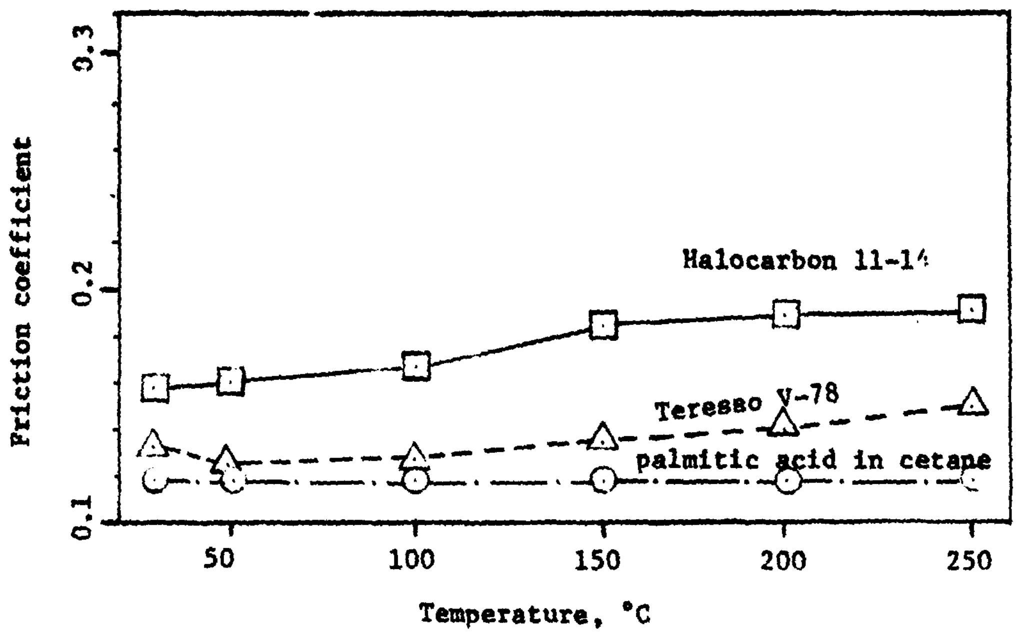


Figure 9. Friction-temperature plots for aluminum oxide, lubricated by Halocarbon 11-14, Teresse V-78, and palmitic acid in cetane.

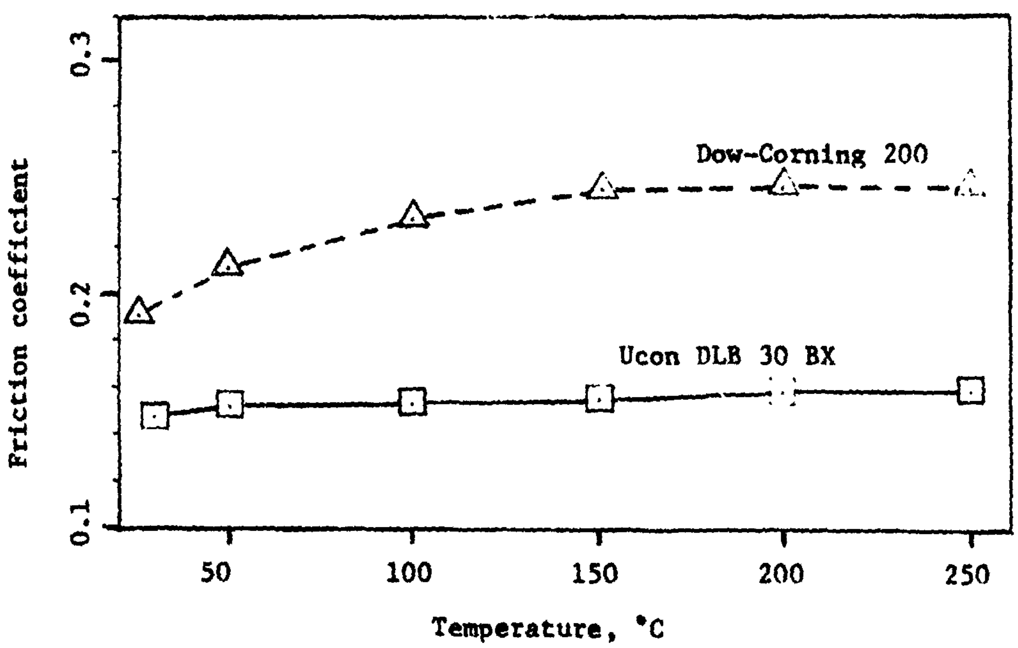


Figure 10. Friction-temperature plots for aluminum oxide, lubricated by Dow-Corning 200 and Ucon DLB 30 BX.

Evaluating the strength of silicon nitride by abrasive wear tests

Introduction

It is clear, from an evaluation of available experimental data, that not all samples of hot-pressed silicon nitride perform equally were in rolling contact applications. Under these circumstances it would be convenient to have a simple non-destructive test which will make it possible to evaluate samples of silicon nitride before undertaking the expense of making them into bearing components.

We have studied the possibility of measuring the resistance of silicon nitride samples to abrasive wear, and using this as a measure of mechanical strength. As is well known, the rate of removal of material by means of a very hard abrasive (in this case silicon carbide), is to a first order of approximation dependent only on the penetration hardness of the material, and in fact is inversely proportional to hardness (1). But second order effects also exist, in that brittle materials show more wear than do ductile materials of the same penetration hardness. Thus a measurement of abrasive wear rate gives information both on the hardness of a material and on its brittleness.

Apparatus

In our tests, after a number of preliminary studies, we finally developed a technique of mounting three silicon nitride specimens in an aluminium disk which fits into one of the rings of a 3-ring Lapmaster lapping machine. During operation of the Lapmaster, each silicon nitride specimen carries the same load and sees the same abrasive, consequently specimens with the same mechanical properties should show essentially the same wear rate. In our tests, each specimen carried a load of 0.33 kg.

Results

In initial tests, three cylinders cut from the same silicon nitride rod were tested. The weight losses after four hours in the Lapmaster were as follows.

Table 1. Weight loss in the Lapmaster

Material	Weight loss: After 2 hours	After 2 additional hours	Total, 4 hours
Cylinder A	.0168 gm	.0277 gm	.0445 gm
Cylinder B	.0162	.0294	.0456 gm
Cylinder C.	.0158	.0302	.0460 gm

It will be seen that there was much more weight loss in the second two-hour period in the Lapmaster (apparently we used more abrasive in the second period) but the relative wear rates of the three cylinders are uniform to within better than 10 percent.

These results encouraged us to apply this technique to silicon nitride specimens of varied, but known, mechanical properties. Mr. H. R. Baumgartner of the Norton Company kindly supplied us with sets of specimens which were the broken ends of bars used in bend strength determinations (and for each specimen supplied a mean bend strength value). Seven abrasive tests of three specimens each were run, and the average abrasive wear coefficients for each specimen was computed.

The results were as follows.

Table 2. Comparison of abrasive and bend strength tests

Sample (Norton #)	Abrasive wear coefficient = non-dimensional wear rate	Mean bend strength (Norton)
438 750	58×10^{-3}	124.9 ksi
438 751	69	95.9
438 752	83	117.9
438 753	59	110.4
438 754	61	114.2

In comparing these two sets of results, we note that specimen 438 750 gave the lowest abrasive wear rate in our test, and the highest bend strength in the Norton test, and thus may be presumed to be the best material in the sample. Conversely, specimen 438 751 was the worst in the Norton tests, and also gave above average wear in our test, and presumably should not be used to make up bearing components. The other three specimens are intermediate in nature, being fairly good in the abrasive tests and fairly bad in the bend strength tests, or vice versa.

The fact that the abrasive wear testing technique seemed to give results which were reproducible and consistent encouraged us to go ahead and try to resolve the problem of why silicon nitride shows such a high abrasive wear rate, far higher than would be expected of a material with penetration hardness (as measured in ref. 2) of 1500 kg/mm^2.

In order to study the problem we measured the abrasive wear resistance of a sizeable number of materials (nine at a time) testing three specimens in each of the rings of the lapmaster. The materials covered a wide range in hardness from aluminum to silicon nitride (N C 132).

In the first series of tests, we used silicon carbide abrasive (hardness,

2500 kg/mm^2) measured the volume worn away for each material, and plotted the wear volume as a friction of hardness (figure 11). It will be seen that all the points lie close to the theoretical line of slope -1. The silicon nitride point lies distinctly above the line, but nothing very extraordinary.

In a second series of tests, we used as abrasive calcium pyrophosphate (hardness 350 kg/mm^2). In this case the way in which the wear rate varies with material hardness is more complicated, since many of the materials are harder than the abrasive. The theoretical function and the experimental data points lie close to the theoretical line, except the silicon nitride data point, which seems clearly too high. In fact, it seems that the silicon nitride wear rate is that which would be expected of a material of hardness 500 kg/mm^2, rather than the actual 1500 kg/mm^2.

As it happens, the silicon nitride material consists of silicon nitride grains of hardness 2200 kg/mm^2, and a magnesium oxide binder of hardness 500 kg/mm^2. Perhaps in this case the abrasive wears away the magnesium oxide binder, thus causing silicon nitride grains to fall out.

Discussion

All in all, the results are encouraging in that it appears that abrasive wear testing can apparently be used as a simple technique for weeding out poor silicon nitride samples. Of course a more valid test of the concept would be to compare abrasive wear results with performance in rolling contacts.rather than performance in another mechanical property test.

Also it seems that abrasive wear testing can shed light on the mechanism of failure of silicon nitride, and thus perhaps make possible the development of stronger formulations.

The use of the Lapmaster as a measuring tool rather than as a manufactur-

ing tool is relatively novel; however, at least one earlier use is described in the literature (3).

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1. E. Rabinowicz, "Friction and Wear of Materials," John Wiley and Sons, New York, 1965, Chapter 7.
2. D. J. Godfrey and P. G. Taylor, "Inorganic non-metallic bearings with special reference to the use of silicon nitride," pp. 265-274 of "Special Ceramics 4," Brit. Ceramic Res. Assn.
3. C. Pritchard, "Role of the lubricant in three-body abrasion," Nature, 226, 446-7, 1970.

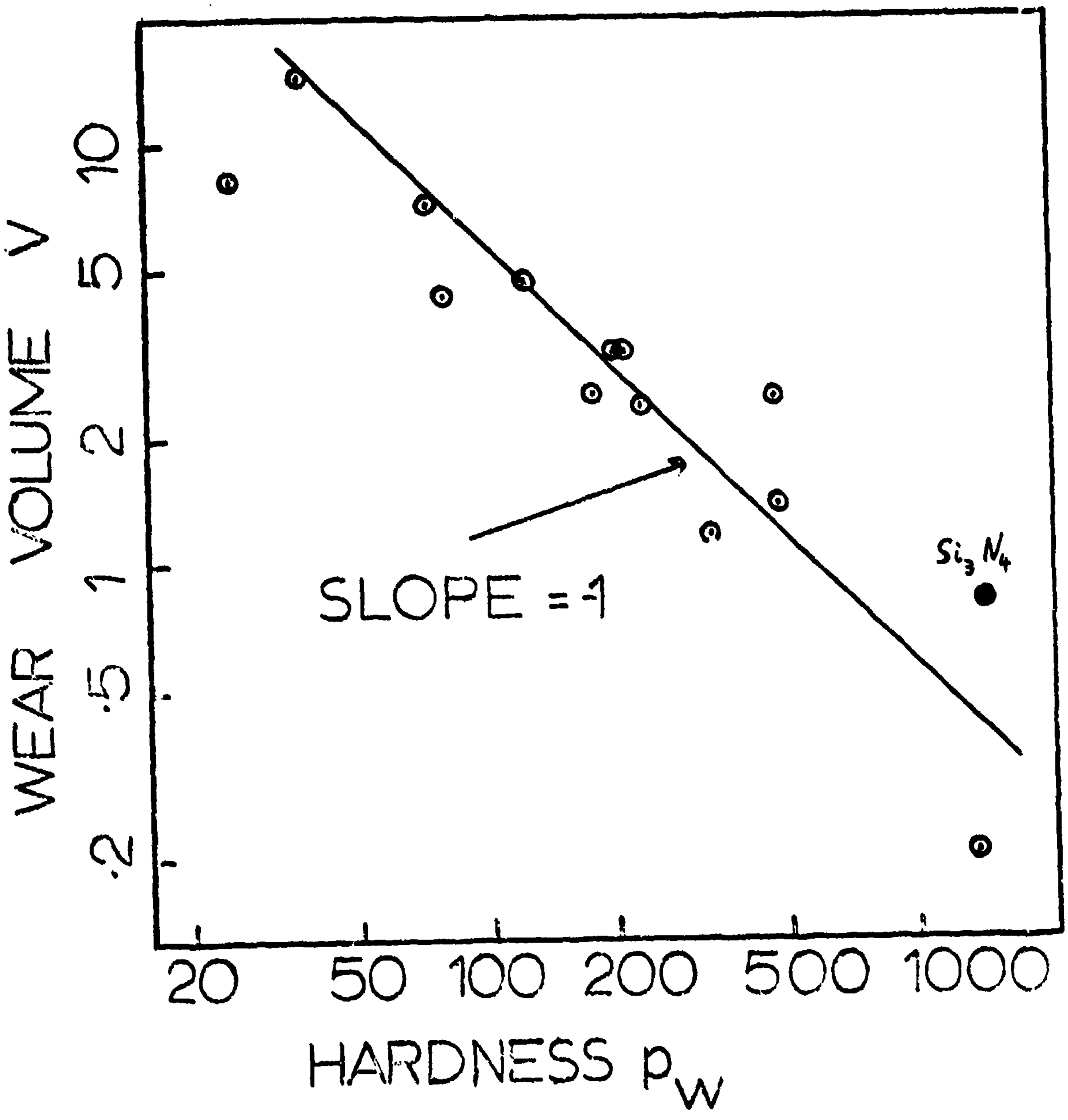

Fig. 11. The relative wear rates for various materials abraded by silicon carbide (hardness 2500 kg/mm^2). The silicon nitride means rather more than would be expected of a material of hardness 1500 kg/mm^2.

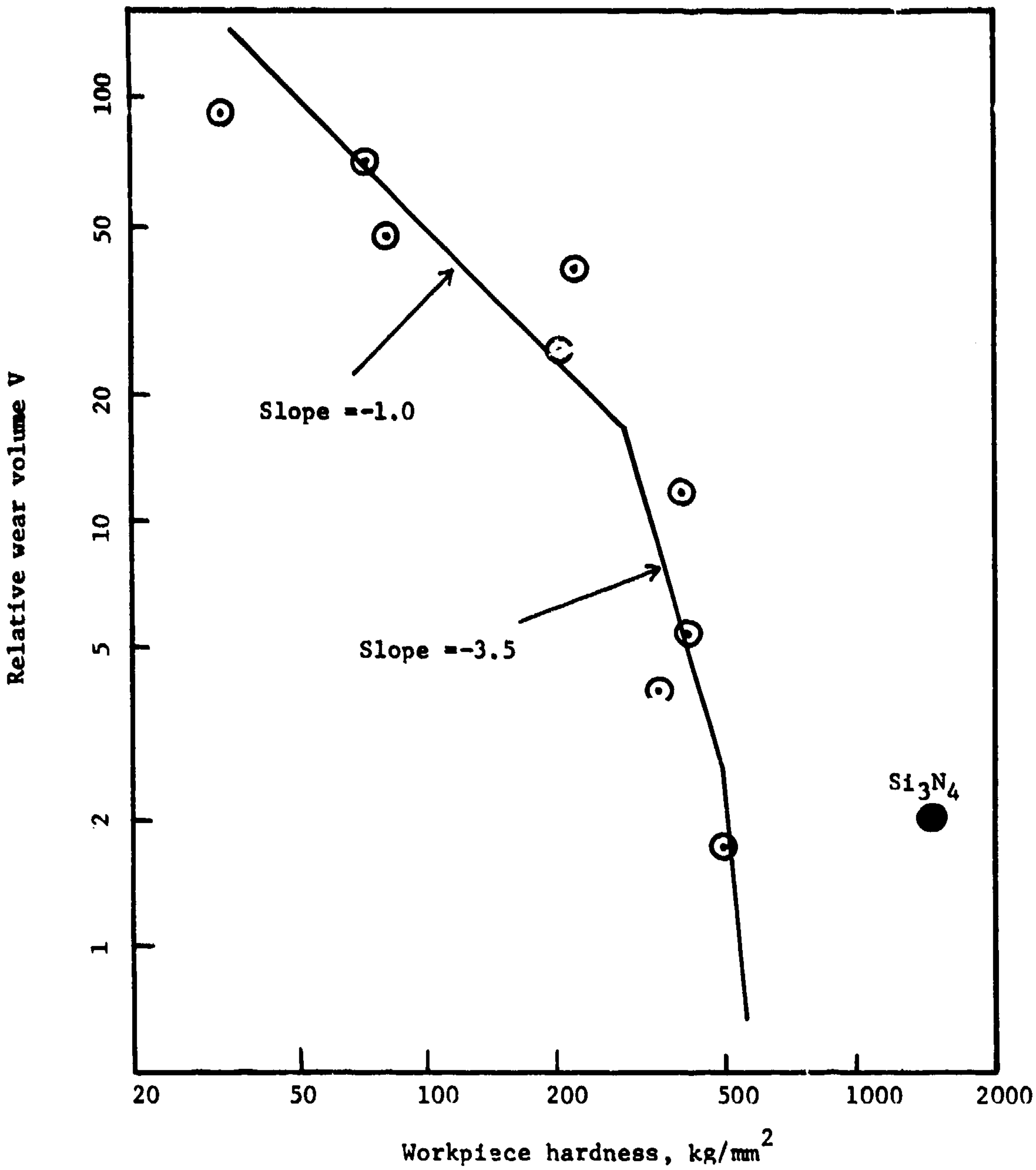

Figure 12. The relative wear rates for various materials abraded by calcium pyrophosphate (hardness 350 kg/mm^2). The silicon nitride wears far more than would be anticipated for a material of hardness 1500 kg/mm^2.

TABLE 1

Test No.	Test Washer	Rolling Element	Lube	Lube Feed Rate (drops/min)	Oper. Temp. (C)	Specific Weight	Viscosity (cs)	Viscosity $N\text{-}\frac{sec}{mm^2}\times10^3$	Load (N)	Max. Hertz Stress (N/mm²)	Calc. film Thickness (h,μm)	Comp. Surface Roughness (σ,μm)	Lube Film Para (h/σ)	Life (10^6 Cycles)	Failure
1	52100	M50(R)	MIL	30	113	0.925	4.0	3.699	4450	1930	0.092	0.13	0.70	1.35	burn up
2	52100	M50(R)	MIL	30	77	0.955	8.3	7.921	4450	1930	0.160	0.13	1.22	3.75	burn up
3	Si_3N_4 (1A)*	M50(R)	MIL	30	96	0.938	5.4	5.061	2225	1520	0.123	0.23	0.54	3.30	RD
4	Si_3N_4 (1A)	M50(R)	MIL	30	110	0.928	4.2	3.896	4450	2100	0.095	0.23	0.40	0.75	RS
5	Si_3N_4 (1A)	M50(R)	MIL	30	121	0.920	3.5	3.219	4450	2100	0.082	0.23	0.37	0.38	RS
6	Si_3N_4 (1A)	M41 (R)	MIL	30	146	0.900	2.5	2.249	4450	2100	0.064	0.23	0.27	1.20	RD
7	Si_3N_4 (2A)	M41 (R)	MIL	30	135	0.913	2.8	2.555	4450	2100	0.070	0.23	0.31	2.20	RS
8	Si_3N_4 (2A)	M41 (R)	DTE	150	107	0.812	7.0	5.683	4450	2100	0.120	0.23	0.52	9.00	RS
9	Si_3N_4 (2A)	M50(R)	DTE	150	105	0.815	7.2	5.867	4450	2100	0.122	0.23	0.52	6.75	RS
10	52100	M41 (R)	DTE	150	113	0.807	6.0	4.841	4450	1930	0.107	0.23	0.46	49.50	T
11	Si_3N_4 (2A)	M50(R)	DTE	150	110	0.810	6.4	5.183	4450	2100	0.113	0.23	0.49	10.50	RS
12	Si_3N_4 (1A)	M50(R)	DTE	150	93	0.823	9.3	7.652	4450	2100	0.146	0.23	0.64	4.20	RD
13	Si_3N_4 (1A)	M50(R)	DTE	150	100	0.818	8.0	6.543	4450	2100	0.131	0.23	0.58	1.10	RD
14	Si_3N_4 (3B)	M50(R)	DTE	150	132	0.796	4.1	3.501	4450	2100	0.085	0.10	0.85	1.20	RD
15	Si_3N_4 (3B)	M50(R)	DTE	150	107	0.812	7.0	5.683	4450	2100	0.134	0.10	1.34	16.20	RS
16	52100	M50(B)	DTE	150	63	0.817	23.0	19.476	2670	3790	0.129	0.10	0.99	21.10	BS, FS
17	Si_3N_4 (3A)	M50(B)	DTE	150	60	0.850	25.0	21.246	2670	4250	0.137	0.10	1.37	60.30	BS
18	Si_3N_4 (3A)	M50(B)	DTE	150	63	0.847	23.0	15.859	3560	4690	0.126	0.10	1.26	12.60	BS
19	Si_3N_4 (3A)	M50(B)	DTE	150	63	0.817	23.0	15.859	3560	4690	0.126	0.10	1.26	1.20	BS, FS
20	Si_3N_4 (4B)	M50(B)	DTE	150	63	0.817	23.0	15.859	3560	4690	0.126	0.10	1.26	64.35	BS
21	Si_3N_4 (4B)	M50(B)	DTE	150	63	0.817	23.0	15.859	3560	4690	0.126	0.10	1.26	9.75	BS
22	Si_3N_4 (4B)	M50(B)	DTE	150	63	0.847	23.0	15.859	3560	4690	0.126	0.10	1.26	8.40	BS
23	Si_3N_4 (4B)	M50(B)	DTE	150	63	0.817	23.0	15.859	3560	4690	0.126	0.10	1.26	5.70	BS
24	Si_3N_4 (4B)	M50(B)	DTE	150	63	0.817	23.0	15.859	3560	4690	0.126	0.10	1.26	6.15	BS
25	Si_3N_4 (5A)	Si_3N_4 (B)	DTE	150	82	0.832	12.5	10.398	3560	5360	0.079	0.10	0.79	4.20	BS, FS
26	52100	Si_3N_4 (B)	DTE	150	68	0.842	18.5	15.574	3560	4690	0.107	0.13	0.82	2.55	BS, FS
27	Si_3N_4 (6A)	M50(R)	DTE	150	121	0.800	5.3	4.239	4450	2100	0.091	0.05	1.83	8.40	RS
28	Si_3N_4 (7A)	M41 (R)	DTE	150	115	0.805	6.0	4.829	4450	2100	0.107	0.05	2.13	72.00	T
29	Si_3N_4 (7A)	M41	DTE	150	113	0.507	6.0	4.841	4450	2100	0.107	0.05	2.13	4.98	RS
30	Si_3N_4 (7A)	M50	DTE	150	110	0.810	6.1	5.187	4450	2100	0.122	0.05	2.44	13.8	RS
31	Si_3N_4 (7A)	M50	DTE	150	116	0.805	6.0	4.829	4450	2100	0.107	0.05	2.13	0.84	RS
32	Si_3N_4 (7A)	M41	DTE	150	116	0.805	6.0	4.829	4450	2100	0.107	0.05	2.13	11.4	RS
33	Si_3N_4 (7A)	M41	DTE	150	121	0.8	5.3	4.239	4450	2100	0.041	0.05	1.83	101	T

*Flat washer identification

R: Roller, B: Ball, S: Spalled, D: Distressed, T: Terminated without failure

TABLE 2

Estimates of 70% Confidence Intervals on L_{50} for Flat Washer Test Groups

Group No.	Combination Rolling Element	Combination Lube	Surf. Finish of Flat (mAA)	Sample Size	No. of Failures	Med. L_{50} 10^6 cycles	70% Confidence Interval LCL 10^6 cycles	70% Confidence Interval UCL 10^6 cycles
1	M50(R)	MIL	0.225	3	2	0.55	0.24	3.79
2	M41(R)	MIL	0.225	2	1	2.93	1.00	1736
3	M50(R)	DTE	0.225	4	4	4.81	2.26	8.32
4	M41(R)	DTE	0.225	1	1	9.00	2.00	1237
5	M50(R)	DTE	0.1	2	1	16.58	5.64	9826
6	M50(B)	DTE	0.1	2	2	39.83	10.36	97.2
7	M50(B)	DTE	0.1	6	6	6.0	3.5	9.1
8	M50(R)	DTE	0.05	4	4	6.06	2.85	10.42
9	M41(R)	DTE	0.05	3	1	151.8	71.34	56703

TABLE 3

Theoretical Life of a Contact in a Flat Washer Test

	Contact Type	Rolling Element	Applied Load (N)	Contact Life L_{10} 10^6 cycles	Contact Life L_{50} 10^6 cycles
1.	steel-steel	roller	4450	6.27	31.35
2.	steel-silicon nitride	roller	4450	3.2	15.99
3.	steel-steel	ball	3560	7.35	36.75
4.	steel-silicon nitride	ball	3560	2.65	13.23

TABLE 4 PREDICTED LIFE OF 120MM BORE ANGULAR CONTACT BALL BEARING AT HIGH SPEEDS

A. STEEL BALLS, I.R. CONFORMITY = 0.54, THRUST LOAD = 13,350N, FREE CONTACT ANGLE = 21°

Shaft Speed		Max. Hertz Stress (N/mm²)		Life (hrs.)			Contact Angle (degrees)		Contact Area (mm²)		
10^6 DN	RPM	Outer	Inner	Outer	Inner	Brg.	Outer	Inner	Outer	Inner	Brg.
2	16700	1636	1671	383	382	205	13.4	28.2	2.64	1.27	3.91
3	25000	1926	1639	59	304	51	8.2	30.0	3.66	1.22	4.08
4	33300	2235	1617	12	257	11	5.2	31.5	-	-	-

B. Si_3N_4 BALLS, I.R. CONFORMITY, THRUST LOAD = 13,350N, FREE CONTACT ANGLE = 18.84°

Shaft Speed		Max. Hertz Stress (N/mm²)		Life (hrs.)			Contact Angle (degrees)		Contact Area (mm²)		
10^6 DN	RPM	Outer	Inner	Outer	Inner	Brg.	Outer	Inner	Outer	Inner	Brg.
2	16700	1637	1652	335	233	147	18.9	26.8	1.88	1.35	3.23
3	25000	1782	1596	104	211	74	14.5	30.0	2.23	1.26	3.49
4	33300	1977	1559	31	197	28	10.6	32.4	2.75	1.20	3.95

C. I) STEEL BALLS, I.R. CONFORMITY = 0.54, THRUST LOAD = 4480N, FREE CONTACT ANGLE = 21°
II) Si_3N_4 BALLS, I.R. CONFORMITY = 0.52, THRUST LOAD = 4480N, FREE CONTACT ANGLE = 18.84

	Shaft Speed		Max. Hertz Stress (N/mm²)		Life (hrs.)			Contact Angle (degrees)		Contact Area (mm²)		
	10^6 DN	RPM	Outer	Inner	Outer	Inner	Brg.	Outer	Inner	Outer	Inner	Brg.
I)	3	25000	1810	1143	103	7763	102	3.3	29.5	3.34	0.59	3.83
II)	3	25000	1566	1100	333	6019	322	7.1	30.6	1.72	0.6	2.32

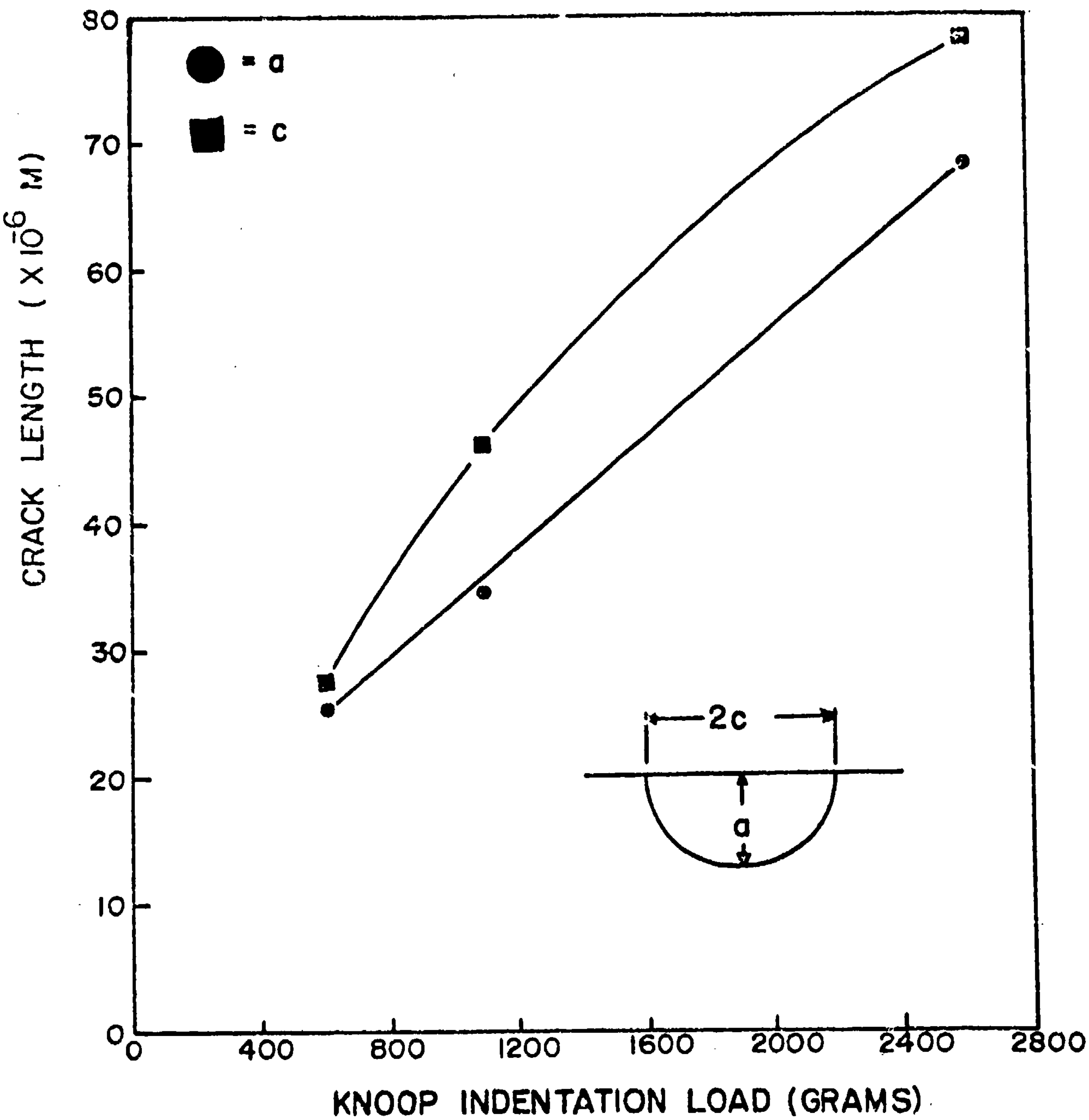

Figure 1. Crack dimensions versus applied Knoop indentation load.

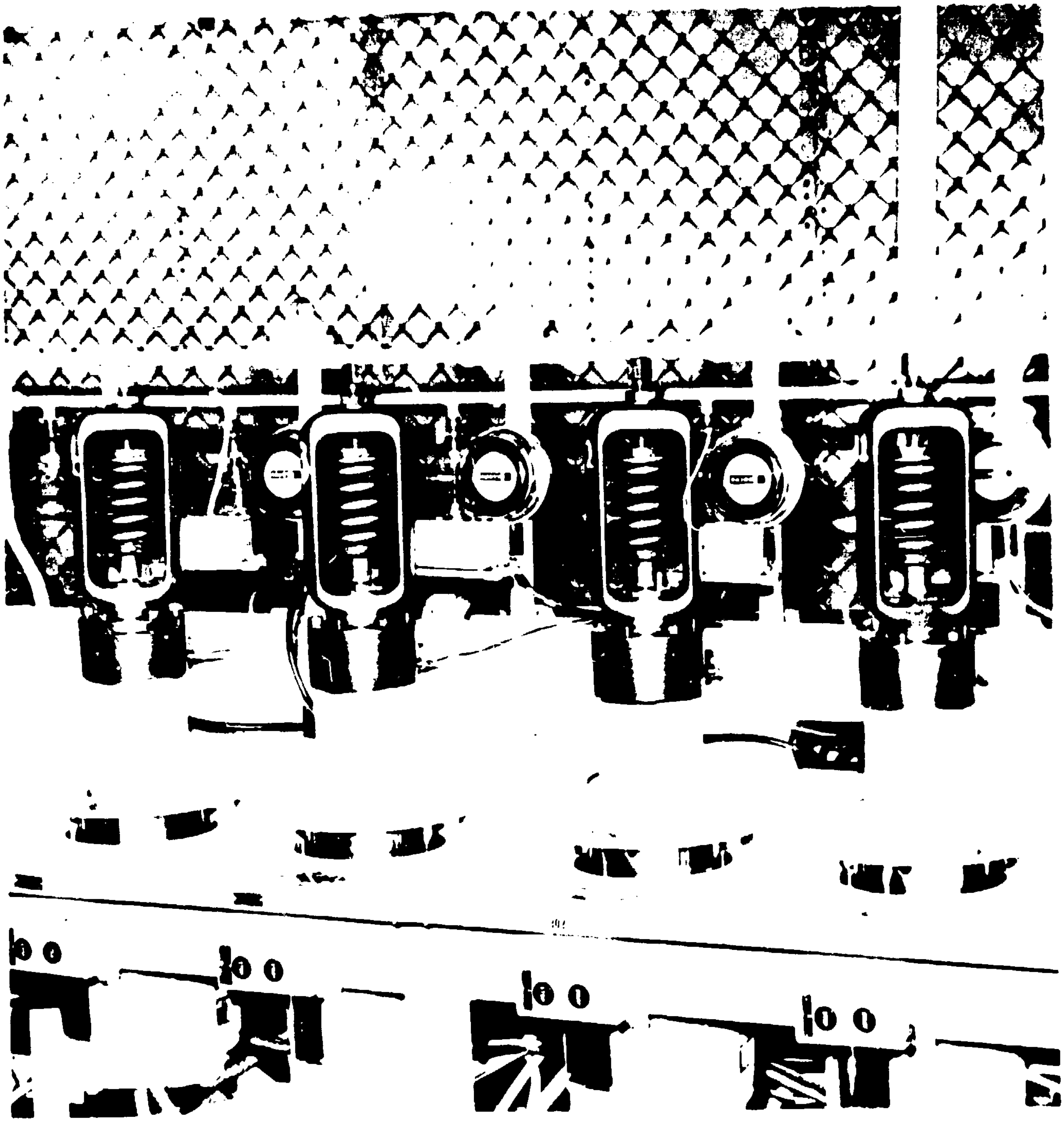

Figure 2. Photograph of a bank of flat washer rolling-contact fatigue testers.

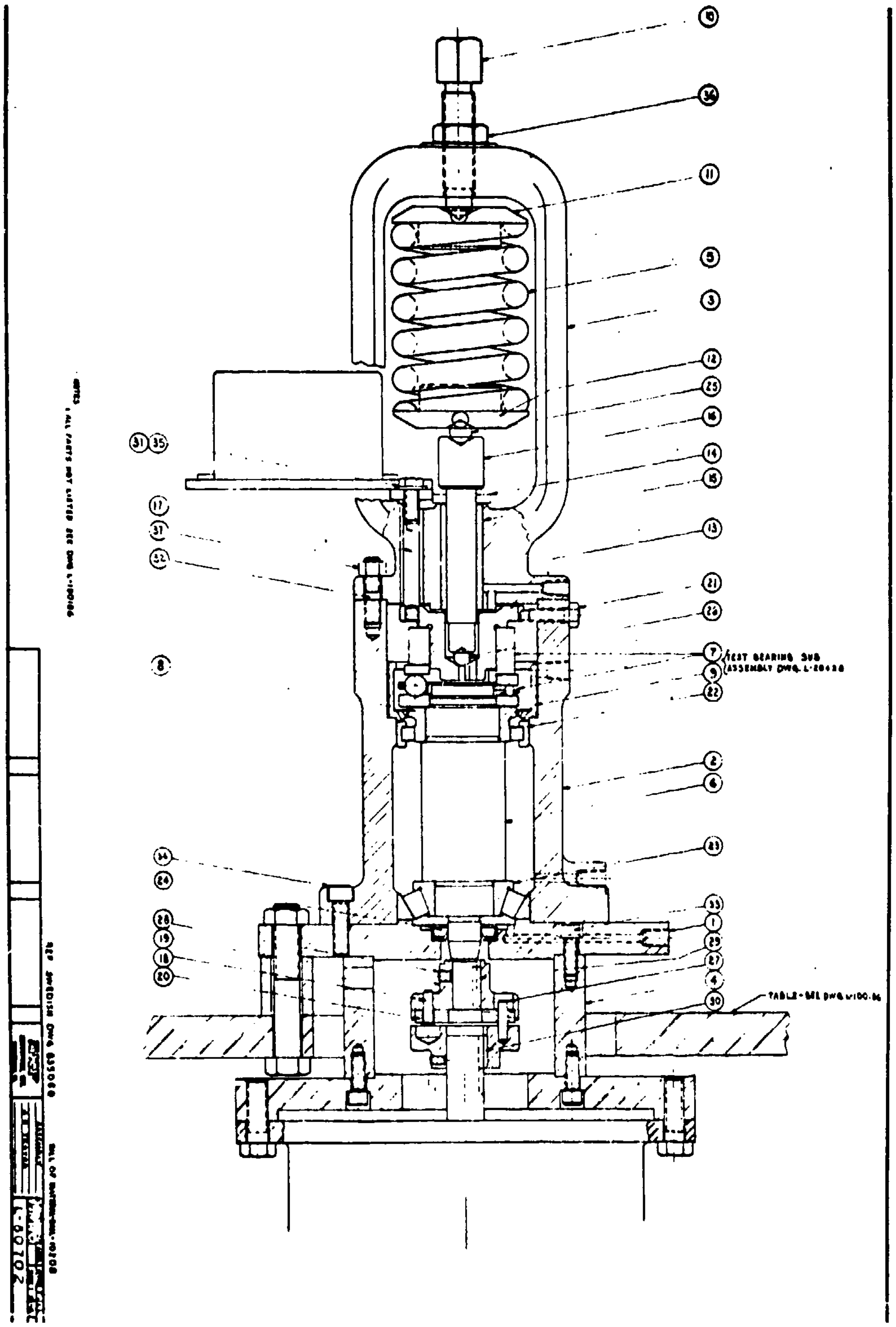

Figure 3. Flat washer tester assembly drawing.

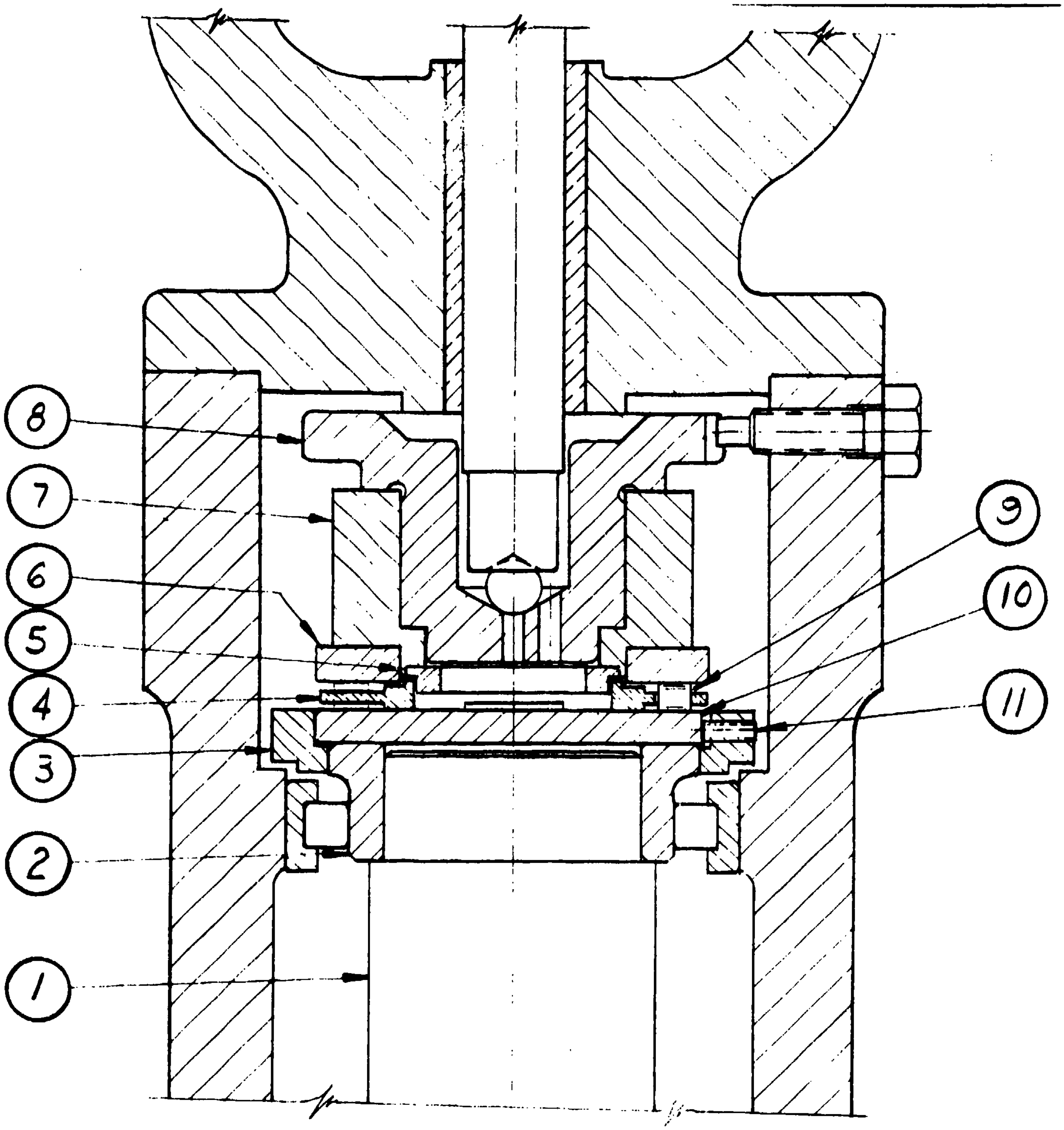

Figure 4. Drawing of test section of flat washer tester showing solid silicon nitride test washer.

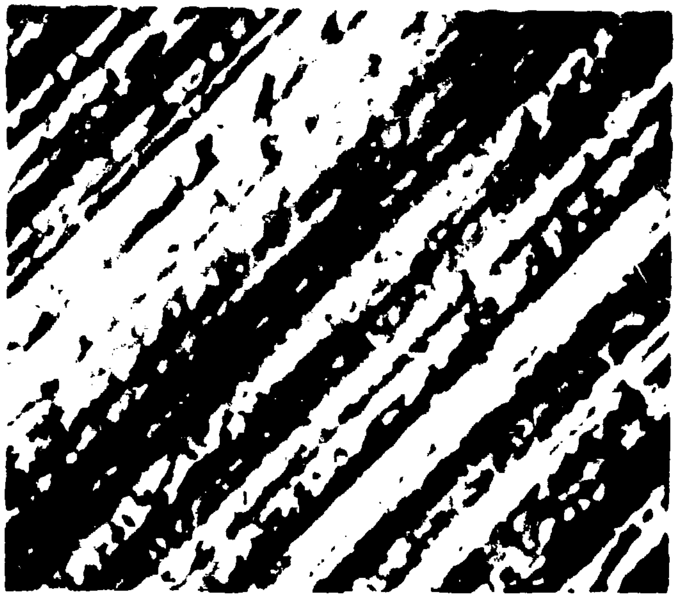

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Figure 5. Surface topography of as-ground silicon nitride flat washer having a surface finish of 0.225 μm AA

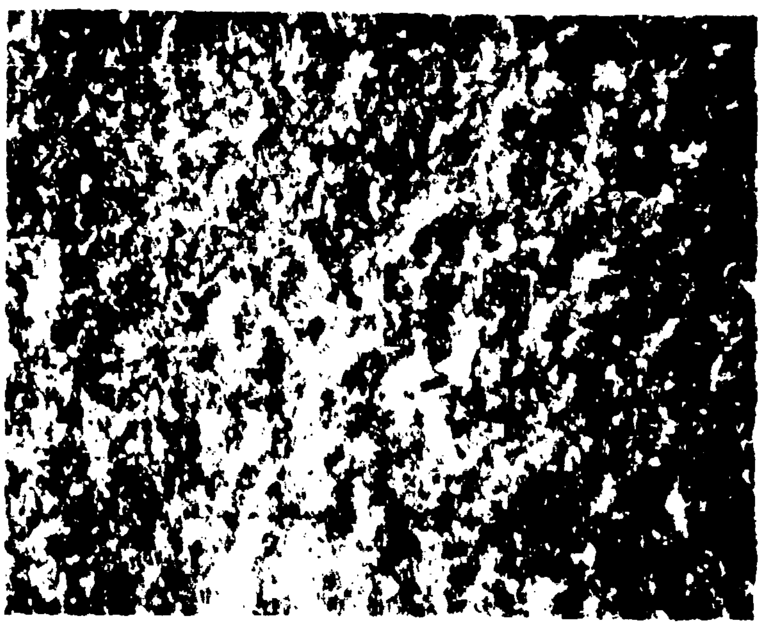

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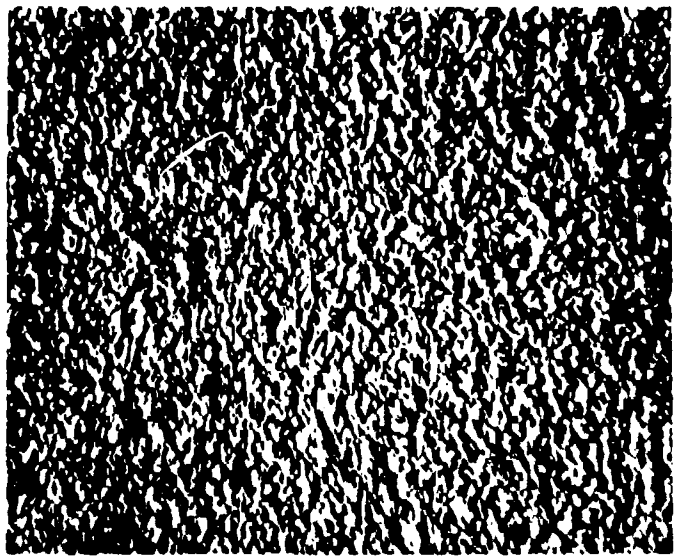
Figure 6. Surface topography of lapped silicon nitride flat washer surface having a surface finish of 0.1 μm AA.



74 327 250X

Figure 7. Surface topography of alumina polished silicon nitride flat washer having a 0.05 μm AA surface finish.

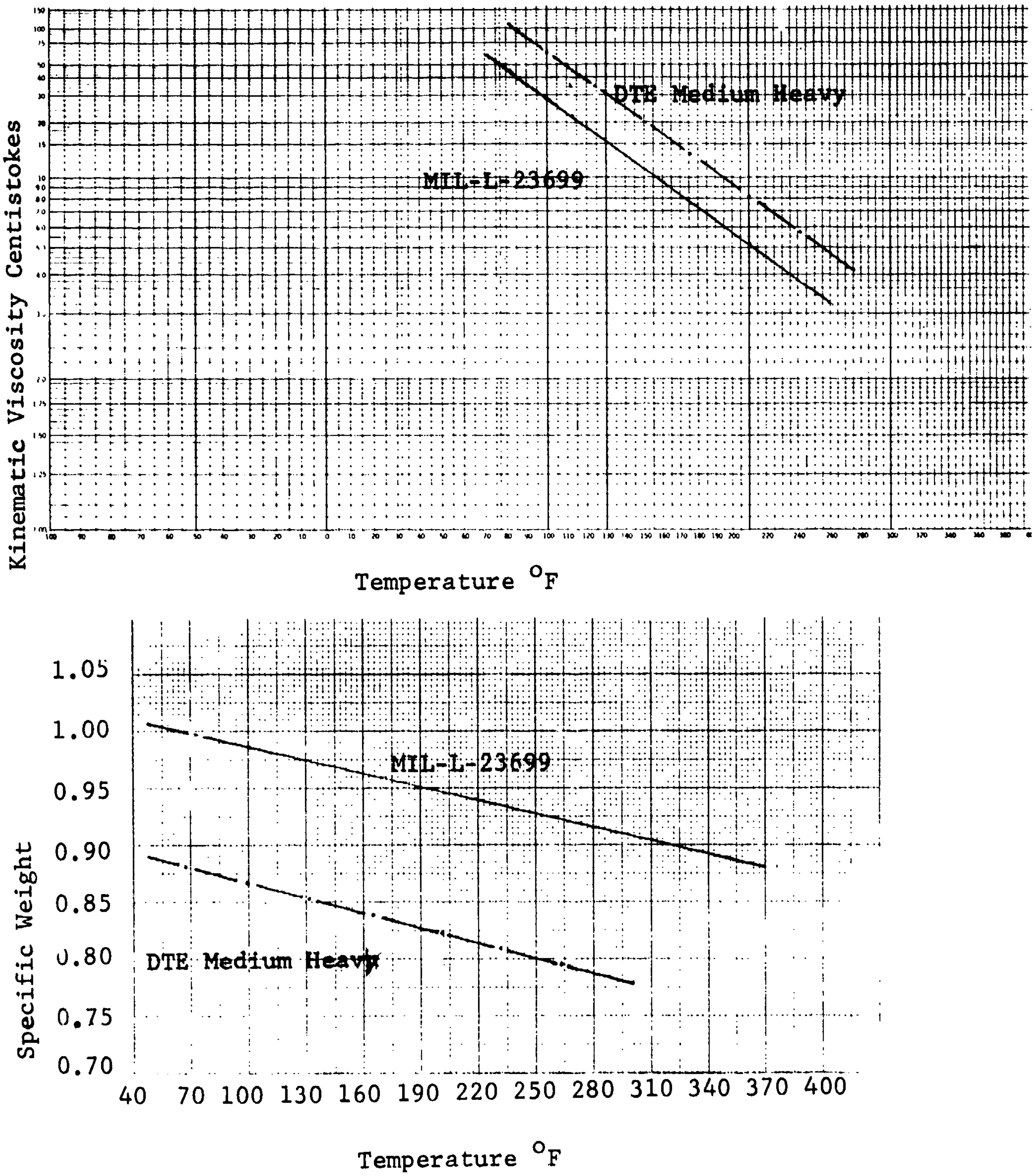


Figure 8. Kinematic viscosity and specific weight of MIL-L-23699 and DTE Medium Heavy Lubricants as a function of temperature.

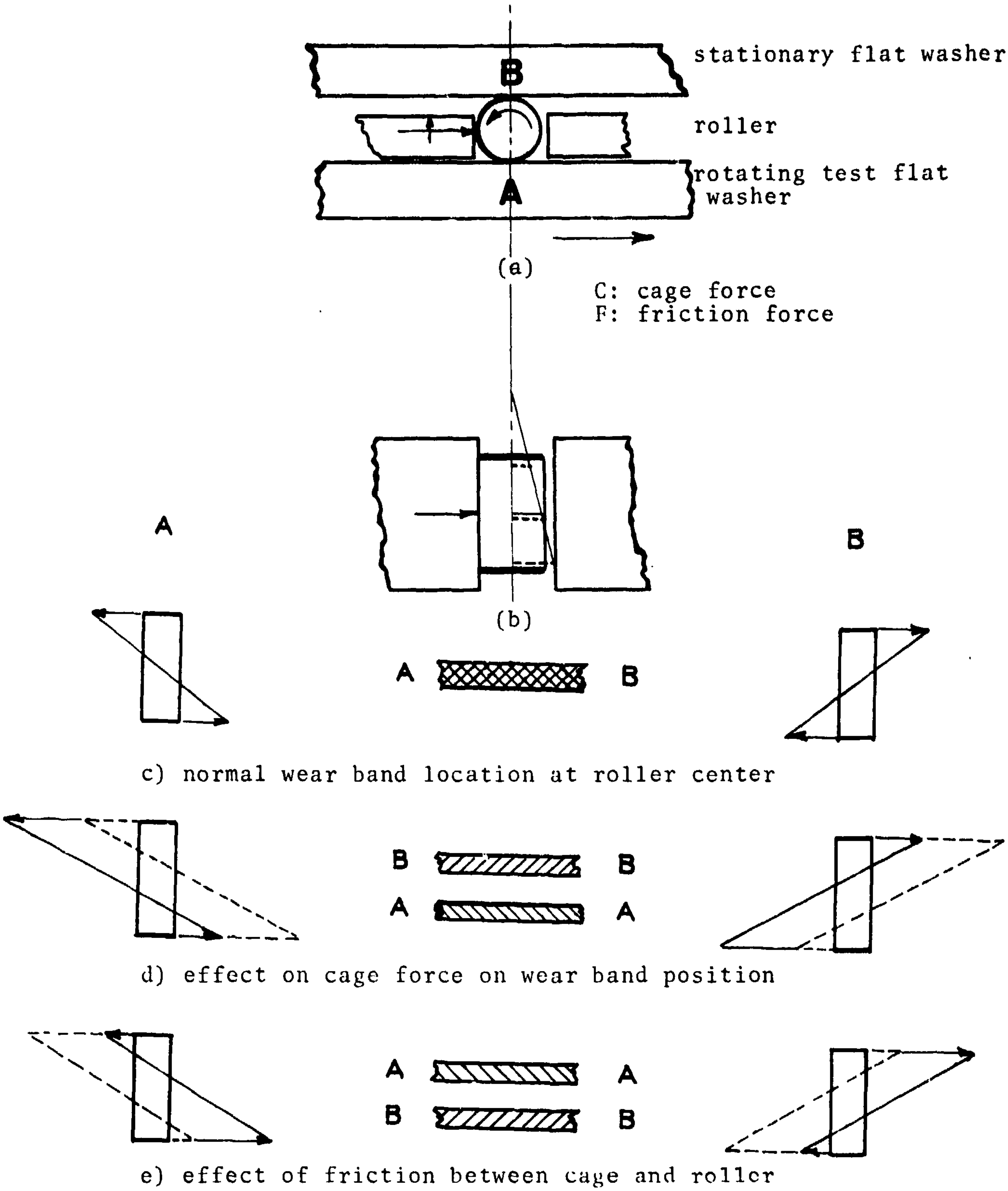
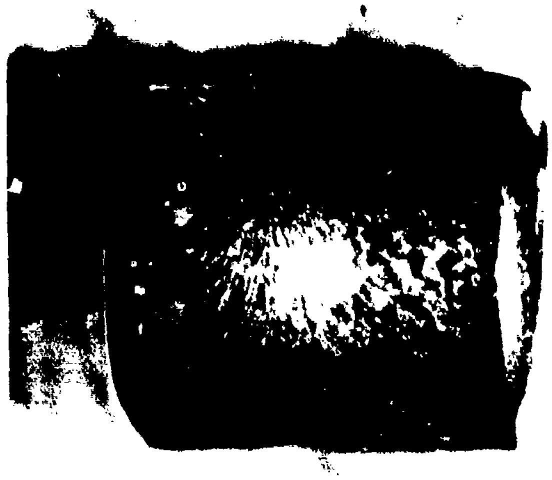


Figure 9. Schematic diagram of roller forces in a flat washer test and their effect on position of wear band location on the roller surface.



74 327 10X

(a)

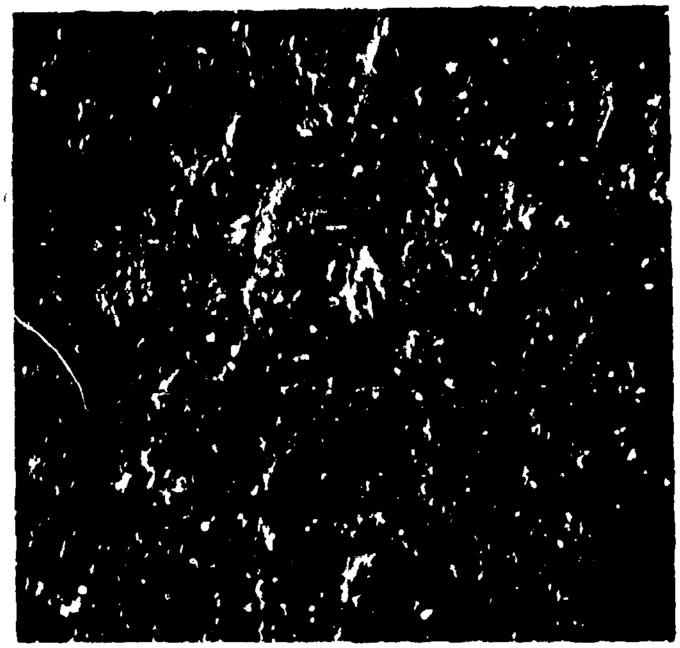
3459 250X

(b)

3462 1250X

(c)

Figure 10. Surface distress on M50 steel rollers run against 0.225 μm AA finish silicon nitride flat washer using MIL-L-23699 lube (Test #3).



1000X

Figure 11. Surface topography of roller track on 0.225 µm AA finish silicon nitride washer after rolling contact.

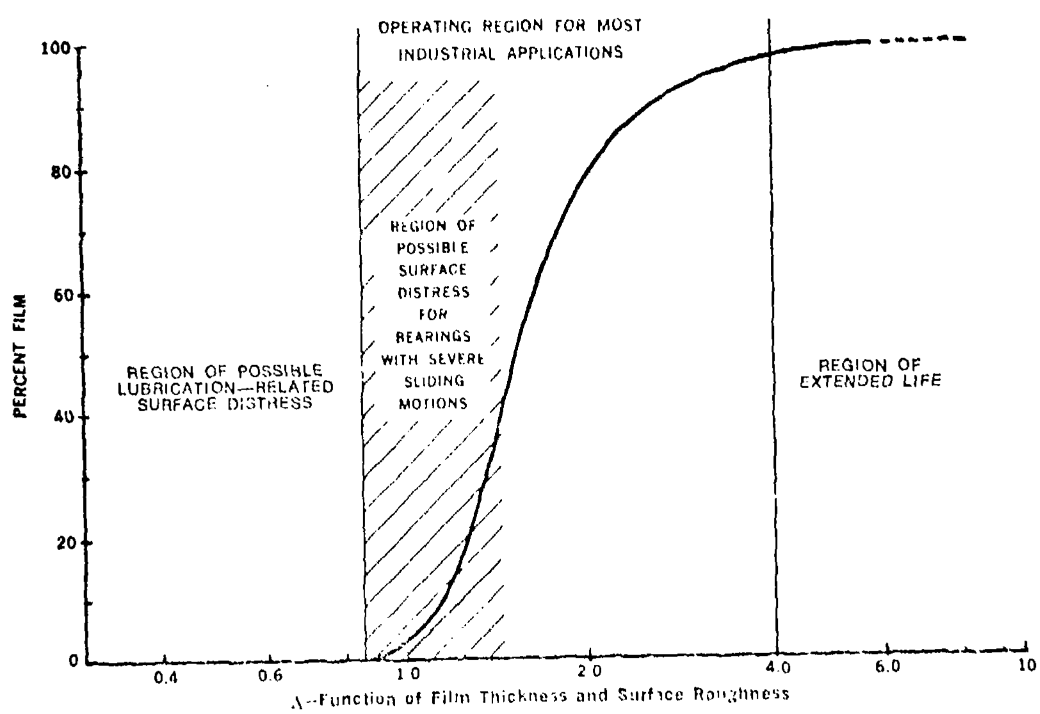
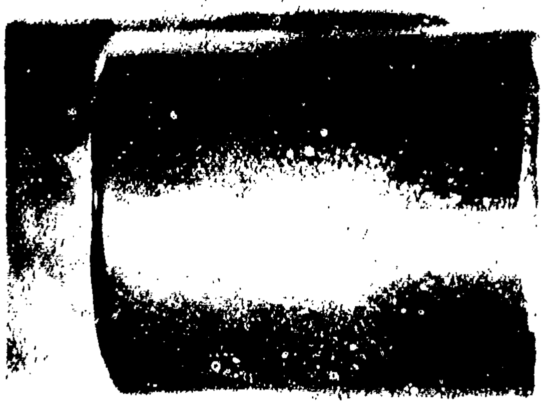


Figure 12. Percent film versus lambda.



74 327 10X

(a)

3458 250X

(b)

3458 2500X

(c)

Figure 13. Surface distress on M41 steel rollers run against 0.225 μm AA finish silicon nitride flat washer using MIL-L-23699 lube (Test #6).

74 327 10X

(a)

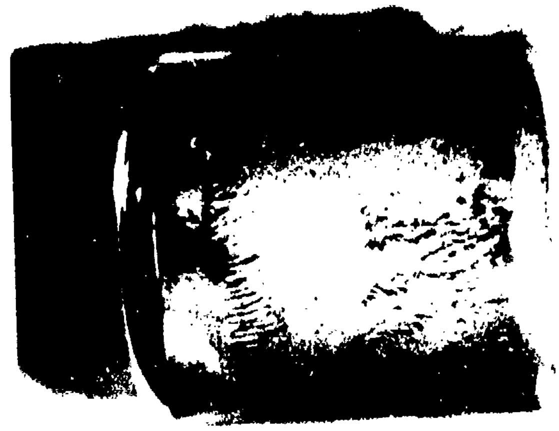
3467 250X

(b)

1250X

(c)

Figure 14. Surface distress on M50 steel rollers run against 0.225 μm AA finish silicon nitride flat washer using DTE Medium Heavy Lube (Test #11).



74 327 10X

(a)

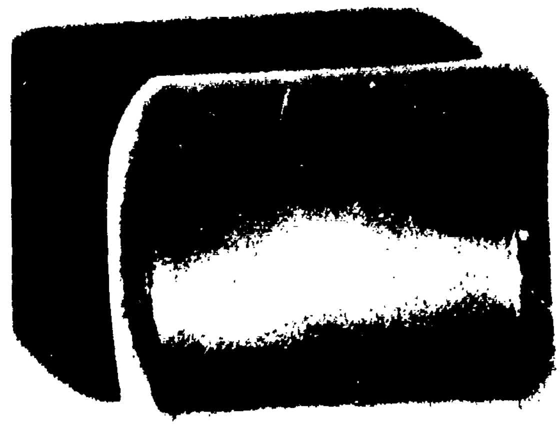
3480 250X

(b)

3481 1250X

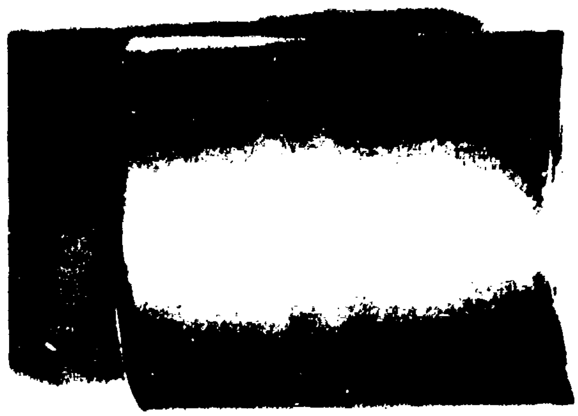
(c)

Figure 15. Surface distress on M50 steel rollers run against 0.1 μm AA finish silicon nitride flat washer using DTE Medium Heavy Lube (Test #14).



74 327 10X

Figure 16. Surface damage on M50 steel rollers run against 0.05 μ m AA finish silicon nitride flat washer using DTE Medium Heavy Lube (Test #27).

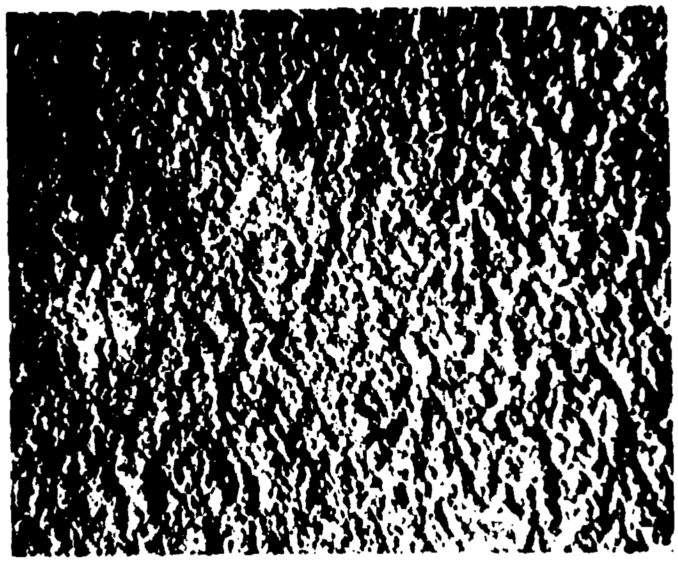


74 327 (a) 10X

3539 100X
(b)

3540 2500X
(c)

Figure 17. Surface damage on M41 steel rollers run against 0.05 μm AA finish silicon nitride flat washer using DTE Medium Heavy Lube (Test #28).



74 327 250X

Figure 18. Appearance of roller track on .05 μm AA silicon nitride flat washer after Test #28.

74 327 10X

Figure 19. Spalled ball run against new 0.1 μm AA finish silicon nitride flat washer using DTE Medium Heavy Lube (Test #17). Note absence of visible wear band.

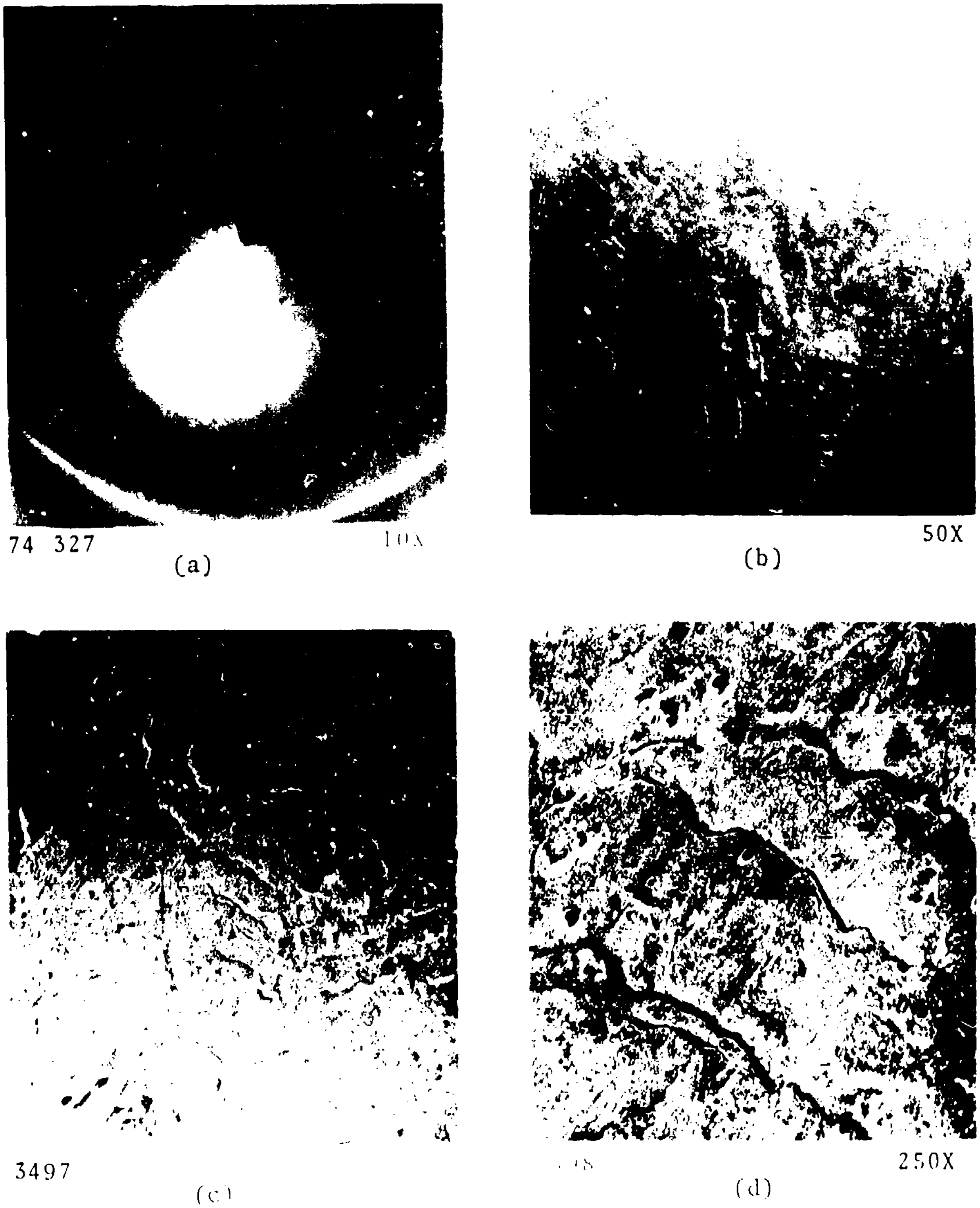


Figure 20. Surface damage in wear track on ball run against grooved silicon nitride flat washer using DTE Medium Heavy Lube (Test #19).

50X

(a)

50X

(b)

Figure 21. Hertz cracking initiated spall found on ball track of silicon nitride flat washer (Test #19).